Waste Heat Recovery Using (s-CO_2) Power Cycle -Applications for Maritime Industry

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Abstract – The predominant source of power in a ship is the diesel engine which has evolved as a highly efficient means of generating necessary power for propulsion and auxiliary uses. However, only less than 50% of the fuel energy is transformed into useful work the rest being losses. It is widely recognized that about 30% of the total energy converted in a Diesel engine is rejected in the exhaust gas. The recently mandated EEDI [1] system for large ships gives credit to ship design for any recoverable energy. While some of the energy saving devices being contemplated, use wind and solar power, it is being recognized that waste heat recovery from the engine exhaust gases and cooling water can still be tapped to generate power resulting in improved energy efficiency of the plant.

One of the ways of recovering heat energy from exhaust gas is to transfer the heat to a medium from which the energy can be recovered. On large ships the medium used is water and steam thus produced is used to heat fuel oil or for electrical energy production through a turbine. In this paper an alternate fluid (supercritical carbon dioxide) is presented as a means for recovering energy through a closed loop gas turbine cycle (Brayton Cycle) It operates significantly at lower temperatures and is non-corrosive, non-toxic, non-flammable and thermally stable. In supercritical state, the s-CO_2 has a high density which results in reducing the size of the components such as the turbine. Supercritical CO_2 gas turbine can generate power at a high cycle thermal efficiency even at modest temperatures of 550°C. The cycle can operate at wide range of pressures 20 to 80MPa.

A case study of the amount of energy recoverable from the exhaust gas of a typical engine installed in an offshore supply vessel is presented along with theoretical calculations for the heat carried out by the exhaust gas and extraction of power which could be generated by the supercritical CO_2 gas turbine plant from the engine

Key words: Waste heat, s-CO_2, water, Brayton cycle.

I. INTRODUCTION

Most of the ships today use diesel engines for propulsion and for producing electrical power. The points of heat rejection form diesel engines which are normally considered to have practical potential for waste heat recovery are the exhaust and the jacket coolant. Heat is usually recovered from the exhaust gas of a main propulsion engine of large ocean liners in the form of steam as it is the most preferred medium for fuel and cargo heating including heating required for domestic services. Heat from jacket cooling water is usually recovered in the form of fresh water generation. Waste heat recovery from auxiliary engines, till recently, was not considered economical and practical except in case of large passenger ships or ships operating with Diesel Electric propulsion system. The debate at the IMO and other international forums on the green house effect of emissions from shipping has changed the outlook of many with respect to looking at various options for enhanced waste heat recovery as a means of improving the overall efficiency of the ship .

In small ships such as the offshore supply and support vessels, the luxury of having space for such mechanisms is generally not available. Other ships generally operating in coastal trade do not generally use fuels which require heating due to their trading pattern hence do not recover energy from exhaust systems. It has been estimated that the global consumption of fuel on coastal ships alone is around 80 million tonnes in the year 2007[1]. Out of this the offshore vessel segment alone contributed around 6.6 million tonnes. The fuel consumption from propulsion engines contributed to about 4.5 million tonnes the rest being from Auxiliary engines [2]. It is also well known that the amount of CO_2 released into the atmosphere is approximately 3.1 times the amount of fuel burnt. Therefore any energy recovered from exhaust gas will result in reduced fuel consumption and result in reduced emissions. This itself may be considered as an incentive to develop WHR systems which have high energy outputs with low footprint in terms of size and weight so that they can be installed on ships serving the offshore sector like supply vessels.

One of the means of recovery of heat from exhaust gases is to use a fluid in supercritical range. The use of supercritical fluids is not new in the power generation industry. Most of the thermal power plants use steam in supercritical stage (23.5 to 38 Mpa) to increase the thermal efficiency of the plant.Similarly nuclear power industry also uses the supercritical water-steam for generation of power. It is beleived that next generation (Gen IV) nuclear power plants will use different fluids such as CO_2 in supercrtical stage for recovery of heat[3].

II. NOMENCLATURE

1. s-CO_2- Supercritical carbon dioxide
2. CO_2 - carbon dioxide
3. OSV – Offshore Supply Vessel
4. NIST – National Institute of Standards and Technology
5. T_{out} – Outlet temperature of the heat exchanger
6. c_p – Specific heat
7. k - Specific heat ratio
8. H- Enthalpy
9. P-Pressure
10. T-Temperature
11. WHR-Waste Heat Recovery
12. IMO – International Maritime Organisation

III. PROPERTIES OF SUPERCRITICAL CARBON DIOXIDE s-CO2

A supercritical fluid is any substance at a temperature and pressure above its critical point. The critical point represents the highest temperature and pressure at which the substance can exist as a vapor and liquid in equilibrium. As shown in Fig.1, above its critical point of 304.13 K (30.98°C) and 73.88 bar, carbon dioxide is a supercritical fluid and has properties of both gases and liquids.

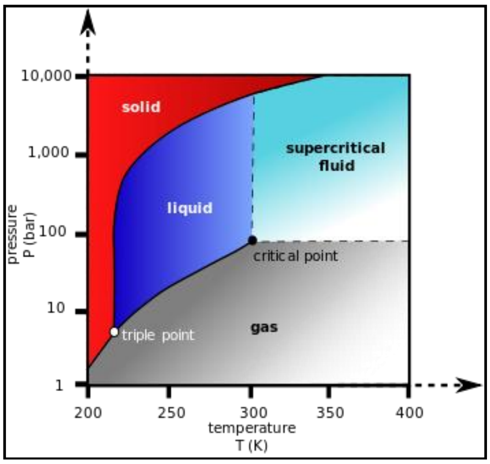

Fig.1: Carbon dioxide- pressure versus temperature (PT) phase diagram [4].

TABLE 1[10]

SUPERCRITICAL PARAMETERS OF H_2O AND CO_2

Fluid	Pressure (bar)	Temp (deg c)	Sp. Heat Cp (kJ/kg-k)	Density ρ (kg/m3)
H_2O	220.75	373.95	459.48	377.03
CO_2	73.88	30.98	71.11	555.1

As shown in table 1, for one degree raise of temperature, CO_2 takes less amount of energy compared to water because of its low specific heat capacity.

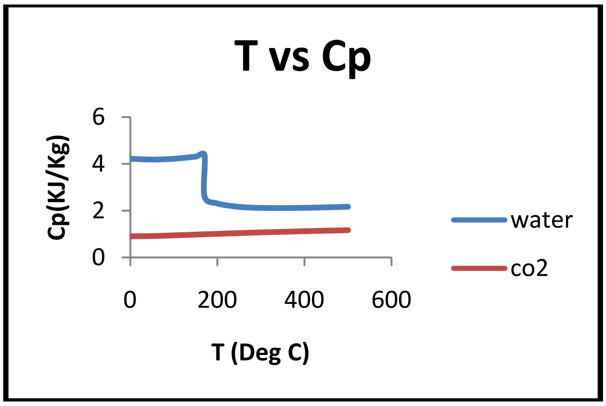

Fig. 2a: T vs Cp

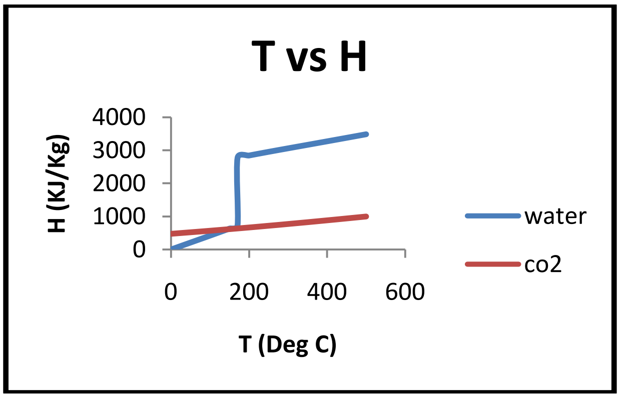

Fig 2b: T vs H

As shown in the fig.2a, at isobaric condition, the specific heat of water is decreasing with increase in its temperature whereas specific heat of CO_2 is increasing slightly but its values are very low compared to water [10].

As shown in the fig.2b, at isobaric condition, enthalpy of both the fluids is increasing. At lower temperatures CO_2 is having more enthalpy compared to water. During phase change water absorbs more energy, where as CO_2 does not have any phase change. After the phase change, the rate of change of enthalpy of water and CO_2 are similar [10].

A. Comparison of heat exchange rate for steam and s-CO_2 from the same heat source.

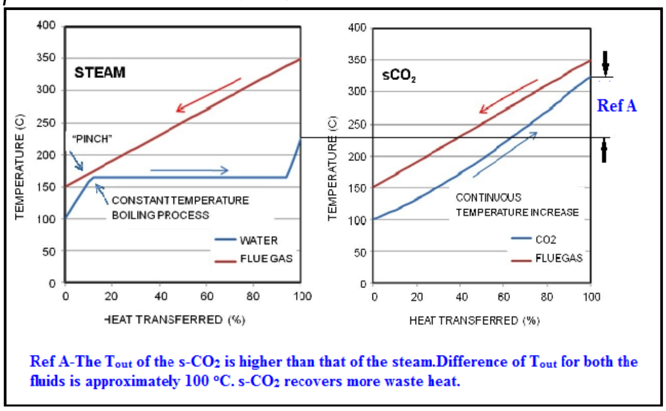

Fig.3: Heat exchange rate for steam and s-CO_2 from the same heat source [7].

As shown in the fig.3, when using water as the medium, pinch point occurs during the constant temperature phase change which limits the maximum fluid temperature. Whereas this phenomenon is not encountered with CO_2, which permits a higher fluid temperature to be achieved for the same heat source. Therefore Carbon dioxide effectively captures waste heat from sources that have an approximately constant heat capacity, such as engine exhaust or other hot gases[6]. This is due to the character of its heat capacity in the supercritical region which provides superior matching to the heat source temperature profile compared to the boiling process utilized with other working fluids such as steam.

B.Other Advantages of CO_2

Carbon dioxide is a clean, non flammable, non reactive, non scaling, non fouling working fluid. and is non corrosive as long as it is maintained in dry condition. A steam based heat recovery system requires multiple stages and multiple pressures to efficiently extract heat from the source. For cyclic operations the steam recovery system requires number of auxiliary equipments and effective management of stresses in the large steam drums which is one of the limiting factors [6]. Compared to organic and steam based waste heat recovery systems, supercritical CO_2 can achieve high efficiencies over wide temperature range of heat sources with components resulting in smaller system footprint [7].

The high density and volumetric heat capacity of CO_2 with respect to other working fluids makes it more energy dense as a result of which, the size of all system components can considerably be reduced without losing performance. As shown in the figure 4 [8], the turbine size and number of stages were reduced when supercritical CO_2 is used as a working fluid. It is estimated that turbines designed for S-CO_2 application can be roughly 1/100th size of steam turbines for the same power output [5]

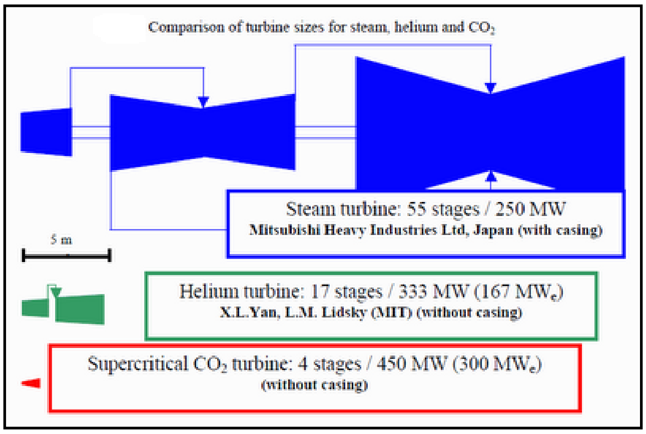

Fig.4: Turbine sizes for different working fluids

IV. EQUATIONS TO DETERMINE THE NET WORK OUTPUT AND EFFICIENCY OF THE IDEAL CYCLE [9]

To make use of this supercritical CO_2 as a working fluid in thermal engine, Brayton closed loop cycle is used as a power cycle as shown in Fig.5.

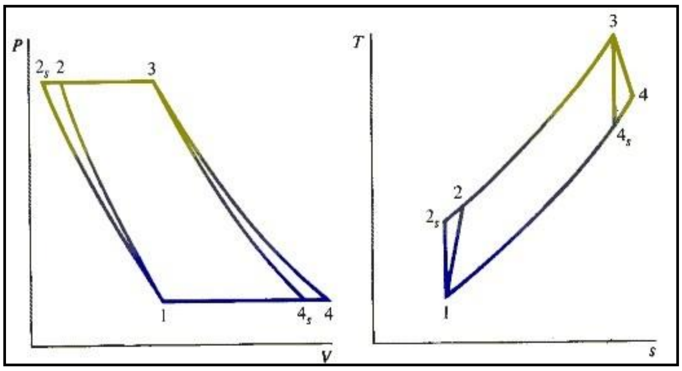

Fig.5. Combined ideal & actual P-V and T-S diagram

Equations for calculating the net work and efficiency of Brayton cycle are as follows:

A) Process 1-2 isentropic compression

1. Out let temperature of the compressor

$$T_2 = T_1 (\frac{P_2}{P_1})^\wedge(k-1)/k) \quad [Here\ k = \frac{c_p}{c_v}] \quad -(1)$$

2. Work input to the compressor per unit mass of the fluid $W_c = H_2 - H_1$ – (2)

B) *Process 2-3 isobaric heat addition*

1. Heat added to unit mass of the fluid

$$Q_{in} = H_3 - H_2 \quad -(3)$$

C) *Process 3-4 isentropic compression*

1. Turbine outlet temperature

$$T_4 = T_3 (\frac{P_4}{P_3})^\wedge((k-1)/k) \quad -(4)$$

2. Work output of the turbine

$$W_t = H_3 - H_4 \quad -(5)$$

D) Process 4-1 isobaric heat rejection

1. Heat rejected to unit mass of the fluid

$$Q_{out} = H_4 - H_1 \quad --(6)$$

E) Net work output per unit mass of the fluid is

$$W_{net} = W_t - W_c \quad -(7)$$

F) Thermal efficiency of the cycle

$$\eta_{th} = 1 - (\frac{Q_{out}}{Q_{in}}) \quad -(8)$$

G) Back work ratio

$$W_r = W_c/W_t \quad -(9)$$

H) Efficiency of the compressor

$$\eta_c = (H_2 - H_1)/(H_2' - H_1) \quad -(10)$$

I) Efficiency of the turbine

$$\eta_t = (H_3 - H_4')/(H_3 - H_4) \quad -(11)$$

J) Actual work done by the turbine

$$\eta_{ta} = H_3 - H_4' \quad -(12)$$

K) Actual work done by the turbine

$$W_{ta} = \eta_t * (H_3 - H_4) \quad -(13)$$

L) Actual enthalpy of the fluid after turbine

$$H_4' = H_3 - W_{ta} \quad -(14)$$

M) Actual work done by the compressor

$$W_{ca} = H_2' - H_1 \qquad -(15)$$

N) Actual work done by compressor

$$W_{ca} = (H_2 - H_1)/\eta_c \qquad -(16)$$

O) Actual enthalpy of the gas after compressor

$$H_2' = H_1 + W_{ca} \qquad -(17)$$

P) Actual back work ratio

$$W_{ra} = (H_2' - H_1)/(H_3 - H_4') \qquad -(18)$$

Q) Actual heat input to the fluid

$$Q_{in} = H_3 - H_2' \qquad -(19)$$

R) Actual heat rejected from the fluid

$$Q_{outa} = H_4' - H_1 \qquad -(20)$$

S) Actual net work output

$$W_{neta} = W_{ta} - W_{ca} \qquad -(21)$$

T) Actual back work ratio

$$W_{ra} = W_{ca}/W_{ta} \qquad -(22)$$

U) Actual thermal efficiency

$$\eta_{tha} = 1 - ((H_4' - H_1)/(H_3 - H_2')) \qquad -(23)$$

V.CASE STUDY

A case study is attempted to estimate the heat energy taken by exhaust gas and mass flow rate of the carbon dioxide for a 2290 kW main propulsion engine fitted in an offshore supply vessel and also to estimate the energy recovered from exhaust gas.

Power cycle used for the OSV engine exhaust waste heat recovery system is s-CO_2 power cycle. The schematic drawing representing the s-CO_2 cycle is shown in the fig.6

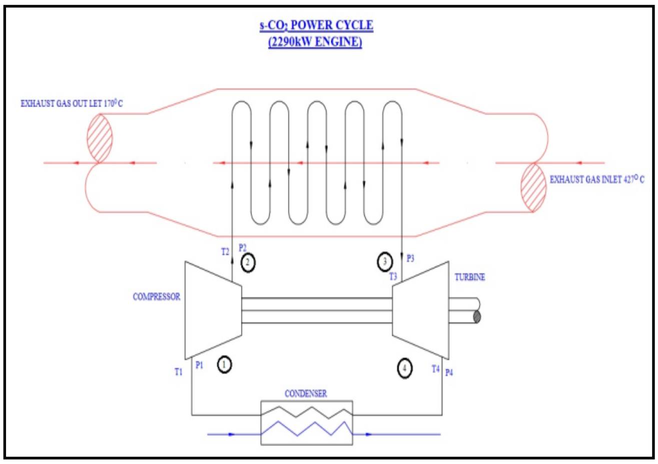

Fig 6: s-CO_2 Power Cycle for OSV Engine (2290 kW)

A. Initial assumptions and properties of carbon dioxide

1. Working fluid = Carbon dioxide

2. Working fluid phase

   a) turbine to compressor = Gas phase
   b) compressor to turbine = Supercritical phase

3. Compressor inlet temperature $T_1 = 36^0$ C (309.15 K)

4. Compressor inlet pressure $P_1 = 14.78$ bar

5. Pressure ratio = 5 (assumed)

6. Compressor inlet pressure P_1 = Turbine outlet pressure P_4

7. Compressor outlet pressure P_2= Turbine inlet pressure P_3

8. Compression and expansion are isentropic processes

9. Heat addition and rejection are at constant pressure

10. Mass flow rate through compressor = Mass flow rate through turbine

11. Specific heat ratio of carbon dioxide k= 1.28 [12]

12. Enthalpy values at different temperatures are taken from published data [10]

13. Compressor efficiency $\eta_c = 0.9$

14. Turbine efficiency $\eta_t = 0.85$

15.Heat exchange rate between flue gas and s-CO_2 is 100%

16. Heat exchange rate between s-CO_2 and coolant is 100%

17. Exhaust inlet temperature $T_{ein} = 427^0$ C

18. Exhaust outlet temperature $T_{eout} = 170^0$ C

B. Isentropic process (1-2)

1. Compressor out let pressure (P_2) = compressor inlet pressure x pressure ratio
P_2 = 73.9 bar
2. Compressor out let temperature from equation (1)
$T_2 = 439.61$ K
3. Enthalpy of fluid at compressor inlet
(H_1) = 502.77 KJ/Kg
4. Enthalpy of the fluid at the out let of the compressor (H_2) = 603.90 KJ/Kg
5. Work done by the compressor from equation (2)
W_c = 101.13 KJ/Kg
6. Actual work done by compressor from equation (15) W_{ca} = 112.37 KJ/Kg
7. Actual enthalpy of the gas after compressor from equation (17) H_2' = 615.14 KJ/Kg

C. Isobaric heat addition process (2-3)

1. By taking reference of the graph -1, the s-CO_2 temperature at the inlet of the turbine is

   T_3 = 407 ^{O}C (680.15 K)

2. Enthalpy of the fluid at (T_3) is H_3 = 876.40 KJ/Kg

3. Heat added to unit mass of the fluid from equation (3) Qin = 272.5 KJ/Kg

4. Actual heat input to unit mass of the fluid from equation (19) Q_{ina} = 261.26 KJ/Kg

D. Isentropic expansion process (3-4)

1. Turbine out let temperature from equation (4)

   $T_4 = 478.31$ K

2. Enthalpy of the gas at (T_4) is H_4 = 668.57 KJ/Kg

3. Work done by the turbine per unit mass of the gas from equation (5) W_t = 207.83 KJ/Kg

4. Actual work done by the turbine for unit mass of gas from equation (13) $W_{ta} = 176.66$ KJ/Kg
5. Actual enthalpy of the fluid after turbine from equation (14) H4'= 699.74 KJ/Kg

E. Isobaric heat rejection process (4-1)

1. Heat rejected in the cooler from equation (6) $Q_{out} = 165.8$ KJ/Kg
2. Actual heat rejected in the cooler from equation (20) $Q_{outa} = 196.97$ KJ/Kg

F.Net work done and efficiency of the cycle

1. Net work output of the cycle per unit mass of the fluid from equation (7) $W_{net} = 106.7$ KJ/Kg
2. Actual net work output of the cycle per unit mass of the fluid from equation(21) $W_{neta} = 64.29$ KJ/Kg
3. Back work ratio per unit mass of the fluid from equation (9) $W_r = 0.49$
4. Actual back work ratio per unit mass of the fluid from equation (22) $W_{ra} = 0.64$
5. Thermal efficiency of the cycle from equation (8) $\eta_{th} = 0.39$
6. Actual thermal efficiency of the cycle from equation (23) $\eta_{tha} = 0.25$

VI.HEAT ENERGY CARRIED OUT BY EXHAUST GAS

A. Main engine Performance data taken from GE 12 V 228 engine

1. Break specific fuel consumption(BSFC) – 500.8 ltrs/hr
2. Exhaust gas temperature (T) – 427 ^{O}C
3. Exhaust gas flow (v_{ext}) – 9.4 m^3/sec
4. Engine power output – 2290 kW
5. Intake air temperature – 49 ^{O}C

B. Energy carried away by exhaust gas

1. Calorific value of the diesel (CV) = 43400 KJ/Kg
2. Diesel fuel weight (ρ_{diesel}) = 0.85 Kg/ltrs
3. Specific heat of the exhaust at 427^{O}C is $C_p = 1.008$ KJ/Kg-K
4. Fuel consumption (m_f) = ρ_{diesel} * BSFC = 425.68 Kg/hr
5. Energy supplied to the engine $Q_{sup} = \text{cv x } m_f$ = 18485796 KJ/hr = 5134.94 KJ/sec
6. Density of the exhaust gas [11]

$$\rho_{ext} = {(1.293 \text{ X } 273 \text{ X } 1.015)}/{(273 + T)} \text{ Kg/m}^3$$

$= 0.5118$ Kg/m^3

7. Mass flow rate of the exhaust gas (m_{ext}) = (v_{ext}*ρ_{ext}) Kg/sec = 4.81 Kg/sec
8. Energy taken away by the exhaust gas

$Q_{ext} = m_{ext} \text{ x } C_p \text{ x } \Delta T$ KJ/sec = 1833.07 KJ/sec

9. Percentage of the energy taken away by the exhaust gas $= {Q_{ext}}/{Q_{sup}} \text{ x } 100$

= 35.69%

C. Mass flow rate of the carbon dioxide

1. Heat energy lost by exhaust = heat energy gained by s-CO_2

$$m_{ext} \text{ x } C_p \text{ x } \Delta T = m_{co\,2} \text{ x } \Delta H$$

Here $\Delta H = H_3 - H_2$

$\Delta T = T_{ein} - T_{eout}$

Mass flow rate of the carbon dioxide

$$m_{co\,2} = \left(\frac{m_{ext}\,XC_p X\,\Delta T}{\Delta H}\right)$$

2. Mass flow rate of carbon dioxide without considering any losses

$m_{co2} = 4.49$ Kg/sec

3. Mass flow rate of the carbon dioxide when 0.9 compressor efficiency

$m_{co2a} = 4.69$ Kg/sec

D. Energy recovered from exhaust gas

1. Cycle net work output $W_c = W_{neta} \text{ X } m_{co\,2a}$

$W_c = 301.52$ KJ/sec (Kw)

2. Percentage of the energy recovered from the exhaust gas $= \left({W_c}/{Q_{ext}}\right) \text{X } 100$

= 16.45%

3. Percentage of the recovered energy in terms of fuel oil supplied energy $= \left({W_c}/{Q_{sup}}\right) \text{ X } 100$

= 5.87 %

VII.DISCUSSION & CONCLUSION

A preliminary analysis of WHR process using s-CO_2 has been presented using the case study. For simplification, the effects of flow and efficiency of heat exchange processes has been ignored. The mass flow rate as indicated in the preliminary study is around 5 Kg/Sec which is very attractive and helps in reducing the size of the turbine. Therefore s-CO_2 can be used to recover the heat from Marine Diesel engines and generate sufficient power for auxiliary use. Considerable amount of work is being undertaken in advanced countries on the application of s-CO2 in heat recovery processes especially to improve the efficiency of nuclear power plants and also as a means to recover waste heat. With the advent of hybrid generation concepts in marine power plant practices , micro turbine based power generation using s-CO_2 has the potential to generate power at a lower footprint (mass and space) when compared to WHR plants using steam. This not only has applications in small but high power vessels like OSV's but also in larger commercial and Naval vessels if the technology is proven and found to be economical. Internationally firms have invested heavily in R&D along with governmental support and are developing this technology which has already been patented. In India we need to invest in R&D by synergizing the efforts of Government, Industry and Academia to develop the technology so as to be able to produce our own products and increase our manufacturing capability. The technologies to be developed are

a. Micro turbine, Compressor & Alternators
b. Heat exchangers
c. Materials and manufacturing processes for high pressure applications
d. Seals & Bearings
e. Instrumentation and control

IMU Visakhapatnam campus is interested to participate in any program on the application of s-CO_2 in WHR process.

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