PART - B.

CHAPTER - 3

OTHER STUDIES ON POWER PLANT CYCLES USING REFRIGERANTS.

Chapter - 3.

OTHER STUDIES ON POWER-PLANT CYCLES USING REFRIGERANTS*.

3.1. Abstract:

The prospects of combined power-cycles using refrigerants as bottom fluids have been studied in the earlier chapters. In this chapter, theoretical cycles efficiencies of different power-plant cycles using either only fluorocarbon refrigerants or some other refrigerants under different pressure and temperature conditions have been studied. It has been shown that R 11, R 12, R 22 and R 113 can be used fairly satisfactorily as the sole working media in super critical power plants for generation of power. It has also been shown that Carbon Tetrafluoride (R 14) can be used satisfactorily in Brayton cycle to generate power. Exhaust gases of internal combustion engines, open-cycle gas turbines and coal fired steam power plants can also be economically utilised for the generation of power by using these refrigerants as working fluids in heat-recovery systems.

3.2. Nomenclature:

h	=	enthalpy, Btu, lb.
t	=	temperature, ^{o}F
T_c	=	Critical temperature, ^{o}F

*A paper on this subject has been accepted for publication in the Journal of ASHRAE. A copy of the Editor's letter has been attached at page 181.

p	=	pressure, psia
P_c	=	critical pressure, psia
P_A	=	boiler pressures, psia
P_B	=	condenser pressure, psia
η	=	efficiency.

3.3. Introduction:

Since many years engineers have been investigating the possibilites of using some fluids other than steam as the working media in power plants. Carbon dioxide has been suggested by some as a suitable working fluid under certain conditions. Recently Angelino and Macchi [38] mentioned the possibility of using Carbon Dioxide as a working fluid and they have computed the thermodynamic properties of Carbon Dioxide up to 750°C. Gaffert [39] has discussed the suitability of the following fluids as top fluids for binary cycles:-

(i) Mercury, Hg.
(ii) Diphenyl, $(C_6H_5)_2$
(iii) Diphenyloxide, $(C_6H_5)_2O$
(iv) Aluminium Bromide, (Al_2Br_6)
(v) Zinc Ammonium Chloride, $Zn(NH_3)_2Cl$

As already mentioned earlier in this thesis the use of Freon turbine by utilising the exhaust gases of Gas Turbines was dealt with in detail by the author as early as 1963.

In 1965, Reti [40,41] also discussed the possibility of using refrigerants as bottom fluids by taking heat from the exhaust gases of Gas Turbines. In the same year Hicks[42] also discussed the possibility of using Aqua Ammonia both as a power-fluid and as a refrigerant to pre-chill gas turbine inlet air.

But no detailed investigation appears to ~~be~~ have been done regarding the use of halocarbon refrigerants as independent working fluids in power plants without the help of some other fluids.

The possibility of using a fluoro-carbon gas turbine for road vehicles[43] was published probably for the first time in 1957. As mentioned earlier, Gokhstein[30] in 1958 showed that a fluoro-carbon power plant cycle would be more efficient than a saturated steam cycle in a nuclear power plant. It has been shown by the author in one of his papers[44] (See appendix No.2) that a propane turbine can produce power by taking heat from the exhaust gases of a coal-fired steam power plant.

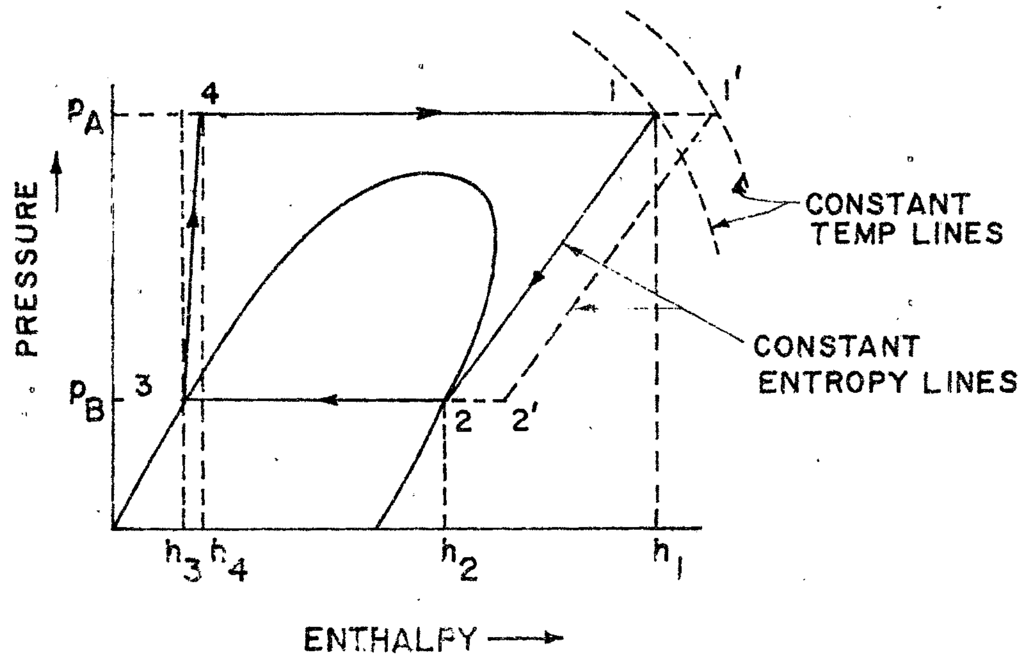
The advantages of fluorocarbon turbine plants, have already been discussed ~~by~~ in chapter 2 (part B) of this thesis. Due to the many advantages listed therein, all possibilities of using these refrigerants as power-fluids in power plant cycles are worth studying in detail. In order to examine the suitability of a new fluid as the working fluid in power plants many factors have to be considered. These are; thermal efficiency, stability, heat transfer properties, transport properties, plant economics and leakage hazards etc. Most of the fluorocarbon refrigerants have been known to be non-toxic and non-flammable. The stabilities, heat transfer properties and transport properties of the fluorocarbon refrigerants have already been determined, to some extent, for refrigeration use. The plant economics can be studied in a realistic manner only after making some pilot plant studies. In this chapter, the only other aspect i.e., thermal efficiency of this type of plant under different pressure and temperature conditions, has been analysed.

3.4. Efficiency calculations using different types of cycles:

A. Super-Critical Cycles:

As all common refrigerants have low critical temperatures and critical pressures, super-critical cycles have to be adopted for Fluorocarbon power plants. A typical pressure enthalpy diagram is shown in Fig.3.1. For some refrigerants like R 113 and R 114, however, it is not possible to draw the isentropic line 1-2 due to the peculiar shape of their p-h diagrams. In such cases and in the case where higher superheat is required, lines similar to 1' - 2' have been taken for calculating efficiency values.

The temperature of condensation has been kept at 100°F throughout to suit average Indian condition of cooling water temperature. For countries like U.S.A. where the cooling water temperature will be lower than 100°F the thermal efficiency will be slightly more than what has been shown ~~in~~ later in this chapter.



SUPER CRITICAL CYCLE ON p-h DIAGRAM

FIG. 3·1

p_A – BOILER PRESSURE p_B – CONDENSER PRESSURE

$h_1 - h_2$ = WORK DONE BY FREON TURBINE

$h_2 - h_3$ = HEAT REJECTED IN CONDENSER

$h_4 - h_3$ = PUMP WORK,

$h_1 - h_4$ = HEAT REQUIRED FOR BOILING

SOME TIMES, LINE 1-2 MAY BE REPLACED BY LINE 1'-2'

The thermal efficiency of a power plant will be the Rankine efficiency, given by:

$$\left.\begin{array}{l}\text{Rankine efficiency without}\\ \text{pump work consideration}\end{array}\right\} = \frac{h_1 - h_2}{h_1 - h_3} \quad \ldots.(3.1)$$

or

$$\left.\begin{array}{l}\text{Rankine efficiency with}\\ \text{pump work consideration}\end{array}\right\} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4} \quad ..(3.2)$$

Suffices 1,2,3 and 4 relate to points in Fig.1:1.

As the pump work cannot be neglected except in cases of low pressures, equation 3.2 has been used for the analyses of super-critical cycles and equation 3.1 has been used for sub-critical cycles. The latest enthalpy values published by the ASHRAE[1] have been used in the calculations. Detailed calculations have been made in case of R 11, R 12, and R 22 as described later and in case of R 113, R 114 and R 500, only one temperature and pressure condition for each has been considered. The maximum pressure and temperature conditions used in the following analyses have been taken from the published ASHRAE pressure enthalpy charts[1] without any extrapolation. At extrapolated conditions, the stabilities of refrigerants being not known, these have not been considered. As per Jordan and Priester[45], halocarbon refrigerants are generally found to be stable upto 800°F. All calculations made below are well within 800°F.

1. Trichlorofluoromethane (R 11):

T_c = 388.4 F, P_c = 640 psia, p_A = 800, 900 & 1000 psia.

p_B = 23.45 psia.

With R 11 as the working substance, efficiencies have been calculated for different turbine inlet temperatures keeping the values of p_A and p_B constant. The results have been plotted in Fig.3.2. From the plot it will be seen that the maximum efficiency with pump work consideration is 29.3 percent when the turbine inlet temperature is 590 F. The Carnot cycle efficiency between the same temperature limits i.e. 590^oF and 100^oF is 46.65 percent. Though the saturated steam cycle efficiency between the same temperature limits will be 38.4 percent it will be unsuitable for steam turbine work, because the dryness fraction of steam after expansion will be 0.655.

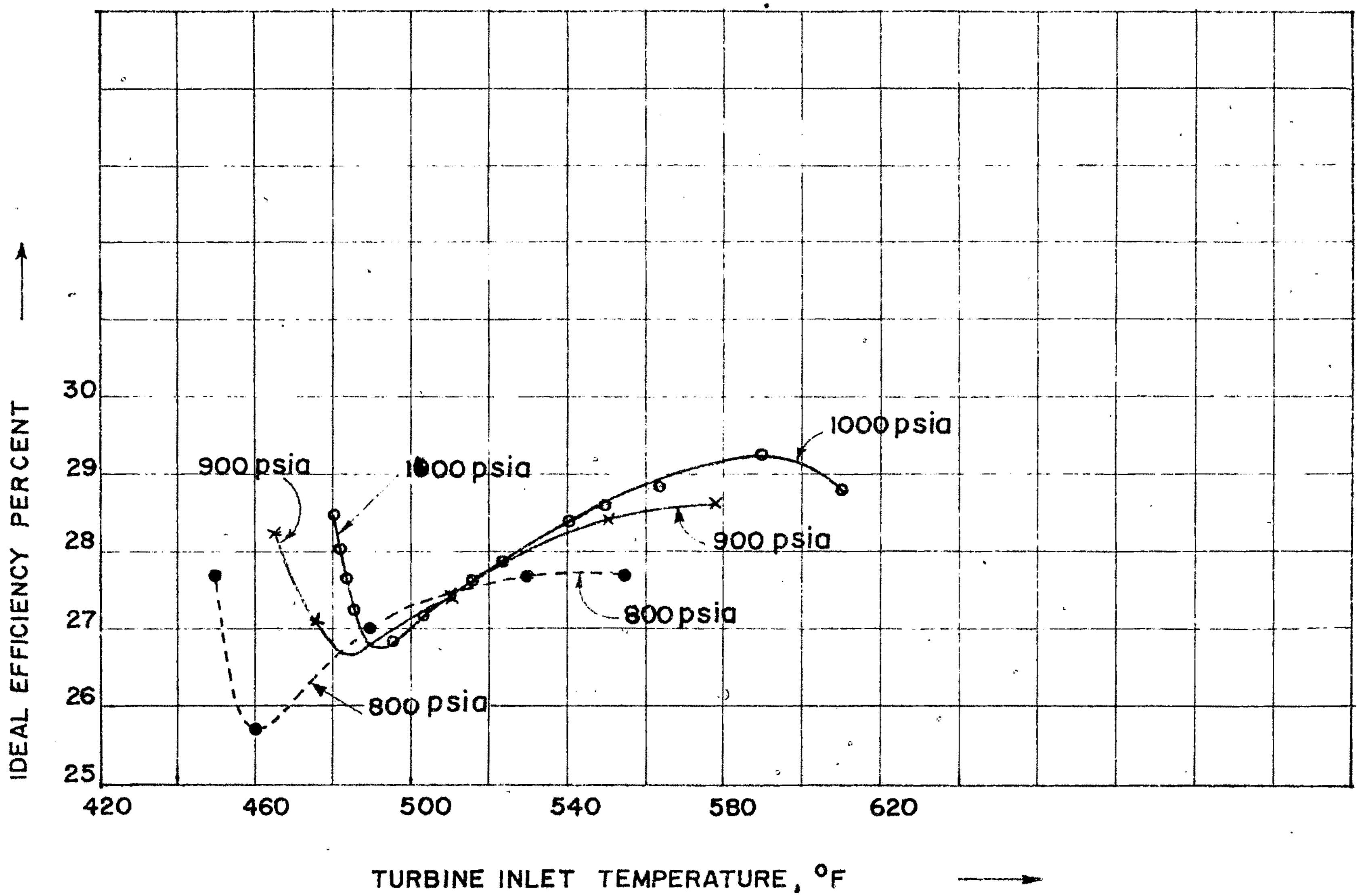
2. Dichlorodiflumro methane (R 12):

T_c = 233.6 F, P_c = 597 psia,

p_A = 1000, 1500, 2000, 3000 & 4000 psia

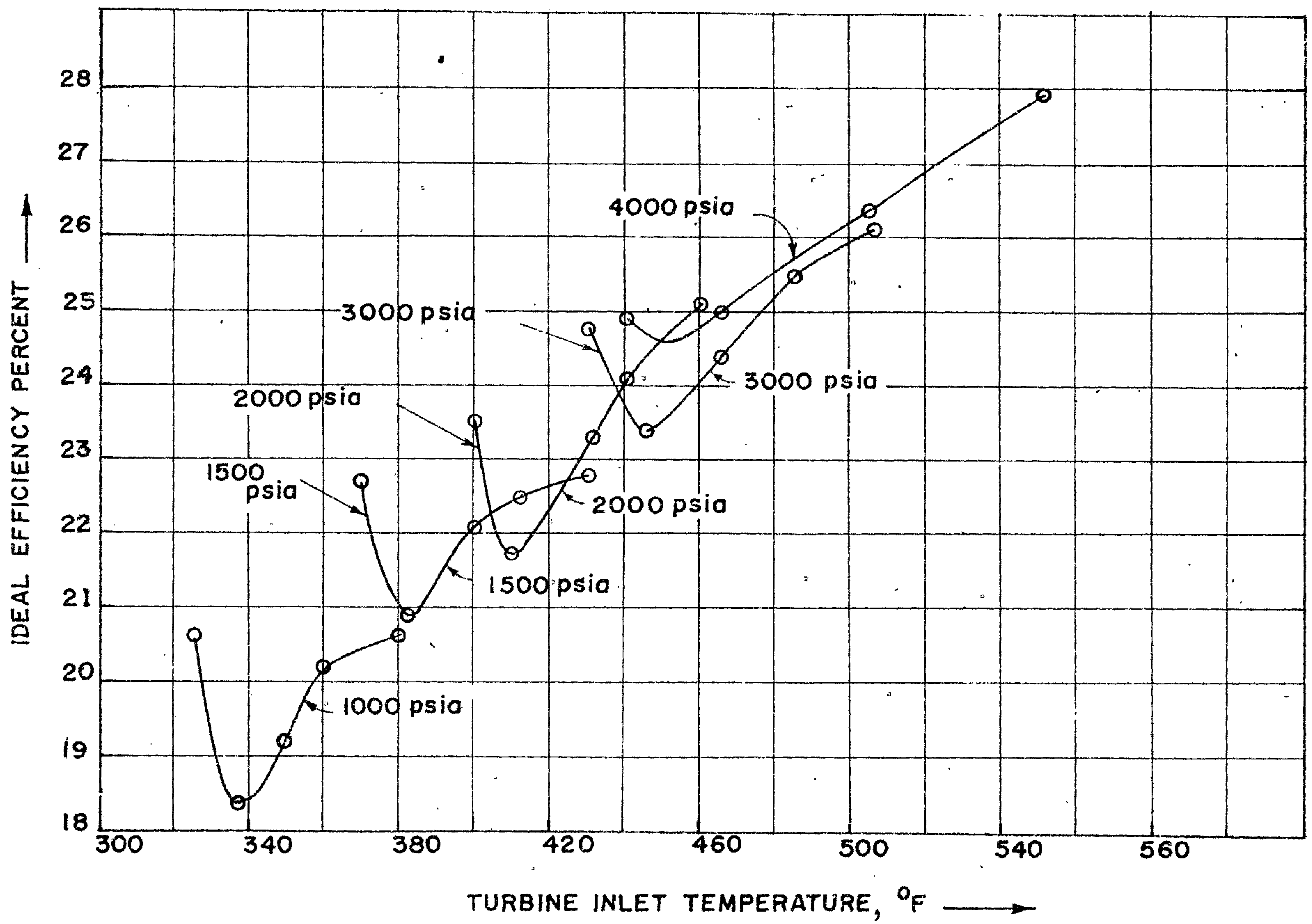
p_B = 131.86 psia.

With R 12 as the working substance, efficiencies have been calculated for different turbine inlet temperatures and pressures keeping the condenser temperature and pressure constant. The results have been plotted in Fig.3.3. From the plot it will be seen that the maximum efficiency with pump work consideration is 28 percent when the turbine inlet temperature is about 550^oF and the pressure is 4000 psia. The Carnot cycle efficiency between the same temperature limits i.e. 550 F and 100 F is 44.5 percent. Though the saturated steam cycle efficiency between the same temperature limits will be 33.5 percent, it will be unsuitable for steam turbine work, because the dryness fraction of the steam after



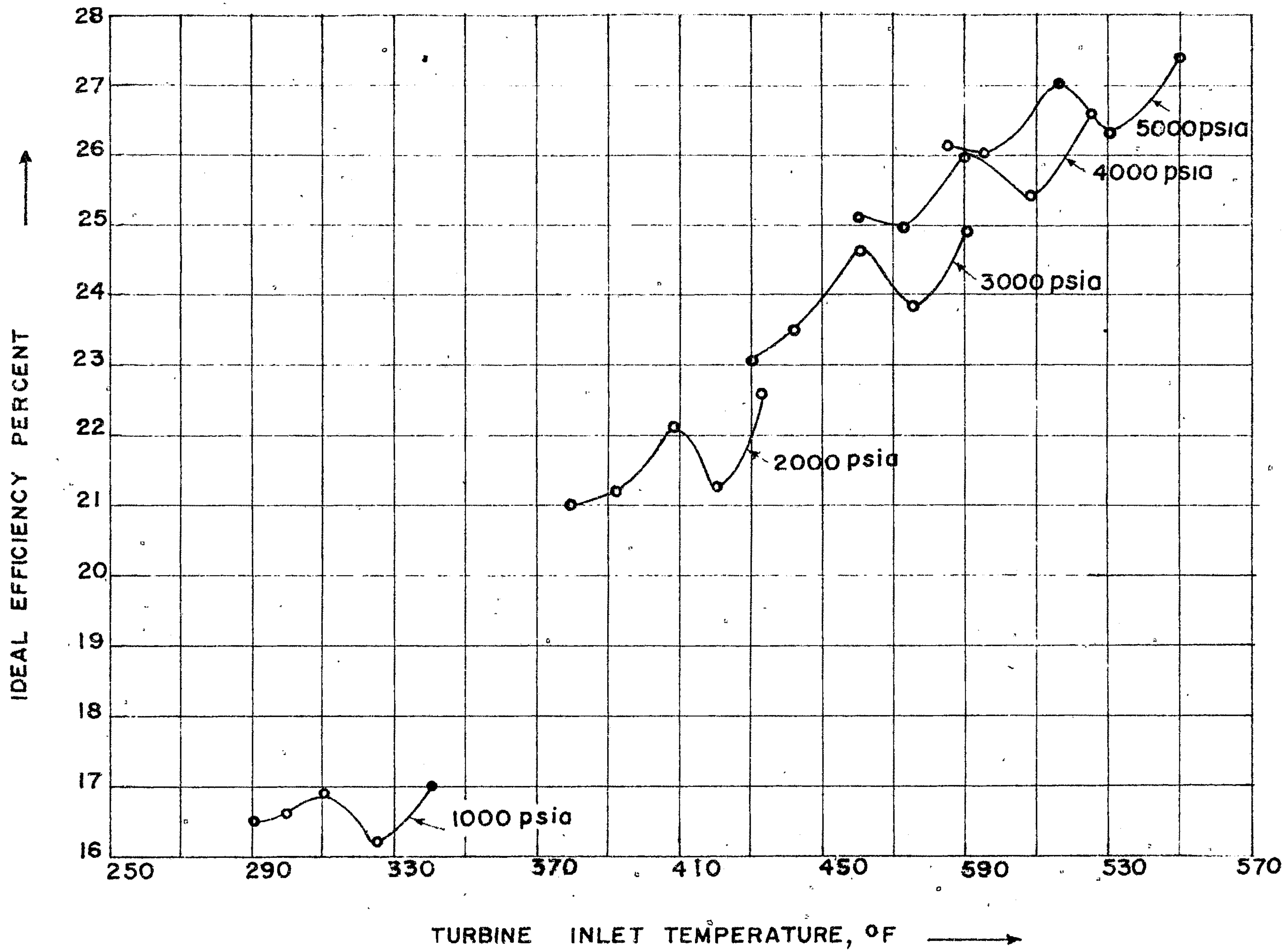
IDEAL EFFICIENCY CURVES OF R-II TURBINE PLANT AT DIFFERENT BOILER PRESSURES

FIG.3·2



IDEAL EFFICIENCY CURVES OF R-12 TURBINE PLANT AT DIFFERENT BOILER PRESSURES.

FIG. 3·3



IDEAL EFFICIENCY CURVES OF R-22 TURBINE PLANT AT DIFFERENT BOILER PRESSURES

FIG. 3·4

expansion will be 0.675.

3. <u>Chlorodifluoromethane (R 22):</u>

T_c = 204.8 F, P_c = 721.9 psia,

p_A = 1000, 2000, 3000, 4000 & 5000 psia.

p_B = 210.6 psia.

A plot similar to the above two has been drawn in Fig.3.4. The maximum efficiency with pump work consideration in this case come to 27.4 percent at the turbine inlet temperature of 550°F and pressure of 5000 psia. The condenser pressure is kept constant at 210.6 psia. The Carnot cycle efficiency between the same temperature limits i.e. 550 F and 100 F is 44.5 percent. Though the saturated steam cycle efficiency between the same temperature limits will be 38.5 percent, it will be unsuitable for steam turbine work, because the dryness fraction of steam after expansion will be 0.675.

4. <u>Other refrigerants:</u>

Out of the remaining refrigerants many cannot be considered for power generation because their critical temperatures are either lower than 100°F or very close to 100°F. Only three refrigerants have been considered whose critical temperatures are sufficiently above 100°F. These values are given in Table No.3.1. Even in these three cases pressure and temperature conditions higher than those mentioned in table No.311 have not been tried due to non-availability of necessary data. The available p-h charts can not be extrapolated as the stability of these refrigerants under extrapolated

Table No. 3.1

Rankine efficiency values of R 113, R 114 and R 500 used in

Supercritical cycles.

Sl No.	Ref.No. with chemical formula	T_c F	p_c psia	p_A psia	p_B psia	t F	h_1 Btu/lb	h_2 Btu/lb	h_3 Btu/lb	h_4-h_3 (Pump work) Btu/lb	Rankine efficieny % as per eqn.(3.2)
1.	R 113 ($C Cl_2F$ - $C ClF_2$)	417.4	498.9	800	10.48	480	136	105	29.0	1.46	28.0
2.	R 114 ($C_2Cl_2F_4$)	294.3	473	800	45.85	360	110	93	31.8	1.53	20.18
3.	R 500 (Azeotrope of R 12 & R 152 (a)	221.9	641.9	800	156.0	272	117.5	103.5	37.0	1.67	15.65

pressure and temperature conditions, is not known.

5. <u>Effect of Lowering of condensation temperature on thermal efficiency:</u>

If the condensation temperature of a R 12 power plant is lowered from 100 F to 40 F, as shown in Fig.3.5, the thermal efficiency of the power plant increases from 24.80 to 28.3 percent (as seen from fig.no.3.3 for 3000 psia and 430°F). But if condenser cooling water is to be cooled by a refrigerating plant driven by the power plant, the net thermal efficiency will be approximately 14 percent as shown below:

From Fig.3.5,

$$\text{Rankine efficiency} = \frac{(113 - 81.4) - 6.33}{(113 - 17.3) - 6.33} = \frac{25.27}{89.37}$$

$$= 28.3\%$$

Heat rejected by the power plant condenser

$$= 81.4 - 17.3 = 64.1 \text{ Btu/lb.}$$

$$\text{Refrigeration effect} = 80 - 31.1 = 48.9 \text{ Btu/lb.}$$

Mass flow of refrigerant required to remove the condenser heat of the power plant cycle $= 64.1/48.9$

$= 1.31$ lb.

Work required by Ref. compressor $= 1.31\ (89.5 - 80)$

$= 12.45$ Btu.

Net output $= 25.27 - 12.45$

$= 12.82$ Btu.

Net thermal efficiency $= 12.82/89.37$

$= 14.35$

Hence it is not desirable to reduce the condensation temperature by refrigeration.

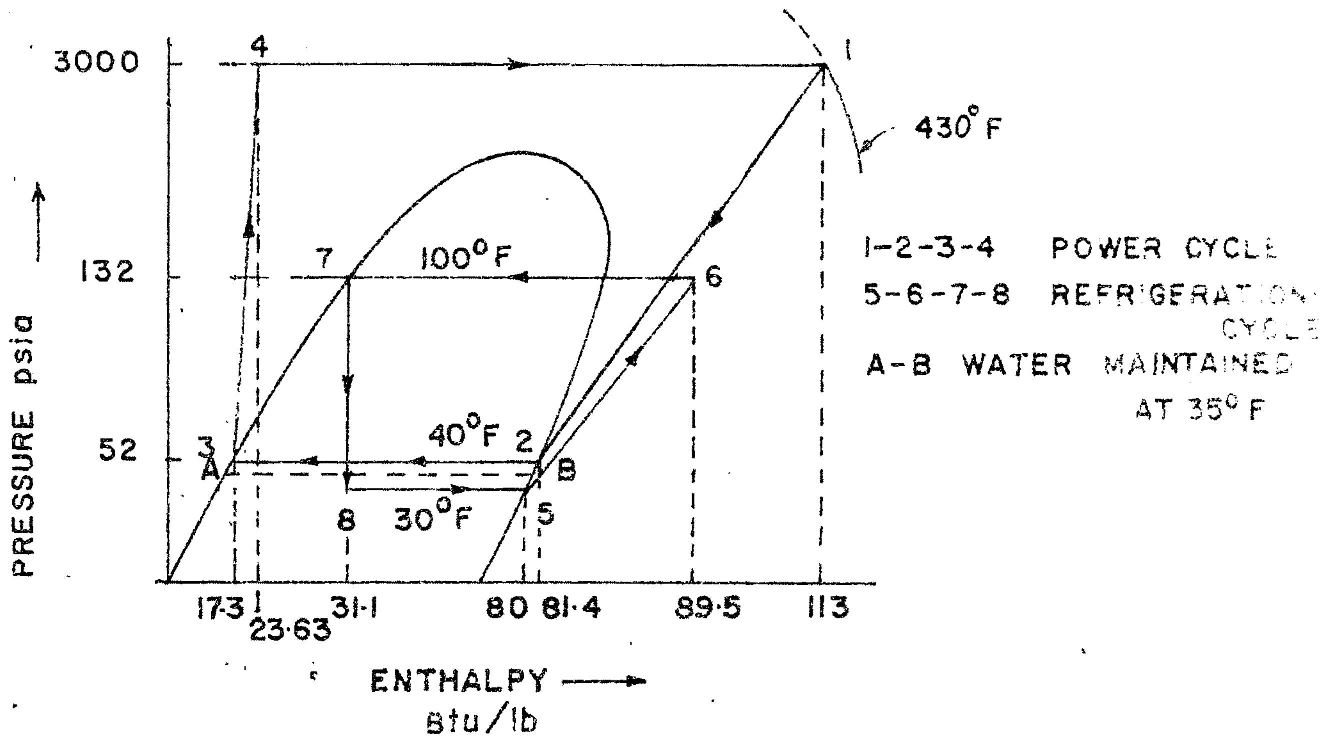


FIG. 3·5 COMBINED POWER AND REFRIGERATION CYCLE OF R 12

B. Sub-Critical Cycles:

Refrigerants, in general, cannot be used as working media in sub-critical power cycles due to low values of critical temperature. Those which can be used give very low thermal efficiencies. Three examples are given below in Table No.3.2.

Table No. 3.2

Thermal efficiencies of sub-critical power-plant cycles using refrigerants as working media.

Sl No.	Refrigerant used as working medium.	Turbine inlet conditions		Condenser Condition		Rankine
		pressure psia	Temp ^oF	pressure psia	temp ^oF	η, %
1.	R 500	511.1	220	155.9	100	13.5
2.	R 216	235.6	320	15.8	100	18.3
3.	Ammonia* R 717	1200	375	211.9	100	18.0

*Enthalpy values of Ammonia have been taken from Ref.23.

C. Brayton Cycle:

In general, refrigerants cannot be heated to very high temperatues for fear of dissociation. It is however possible tou use some refrigerants in Brayton cycle for generation of power, but in these cases the cycle efficiency is found to be lower than gas turbines using air as working medium. As the refrigerants are to be heated indirectly, the overall efficiency will be further reduced. A carbon tetrafluoride (CF_4) plant as shown in Fig.3.6 gives a cycle efficiency of 31.8 percent.

3.5 Discussion of results:

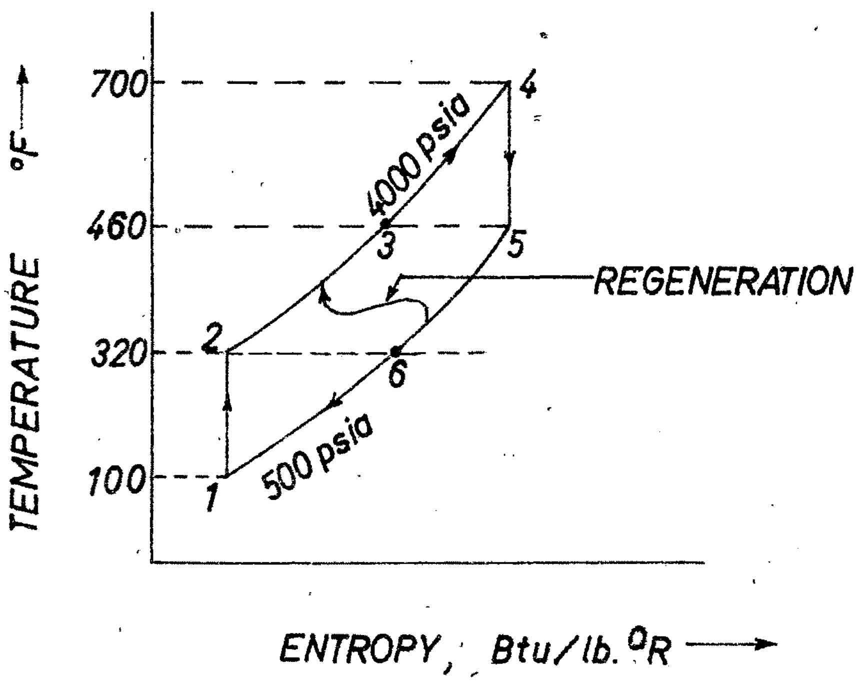
A. Super critical cycles:

(i) Though the Rankine efficiencies of fluorocarbon super-critical cycles are found to be lower than those of the saturated steam cycles operating between the same temperature limits, the overall efficiency of the power plants operating on the former cycle is expected to be more than those operating on the latter cycle, due to lesser temperature difference between cooling flue gases and the boiling refrigerant in a super-critical plant as compared to the isothermal evaporation of steam in a saturated steam plant.

(ii) All other advantages and disadvantages of fluorocarbon power plants as described in chapter 1 (Part B) of this thesis will hold good.

(iii) It will be seen from all the three sets of curves relating to super-critical power plants (figs.3.2,3.3 and 3.4) that the variation of efficiency with temperature is not regular. All the three refrigerants yield without any regularity, a point of lowest efficiency in between the temperature limits considered. This sudden drop in efficiency with increase of temperature requires further investigation.

(iv) It is seen that fluorocarbon refrigerants will have to be compressed to high pressure and heated to high temperature



1–2 COMPRESSION
2–4 HEAT ADDITION
4–5 EXPANSION
5–1 HEAT REJECTION

h_1 =143 Btu/lb
h_2 =175 Btu/lb
h_3 =209 Btu/lb
h_4 =272 Btu/lb
h_5 =220 Btu/lb

FIG.3.6 BRAYTON CYCLE OF CARBON TETRAFLUORIDE.

like steam, inorder to give high thermal efficiencies. Heating to high high température will, of course, have to be limited to the point of dissociation.

(v) Pure refrigerants give higher thermal efficiencies than azeotropes. With same boiler pressure of 800 psia the efficiencies with R 113 and R 114 are 28% and 20.18% respectively, whereas the efficiency for R 500 is 15.65%.

(vi) As the highest temperature considered is about 550°F, exhaust gases of Internal Combustion Engines, open-cycle Gas Turbines and the flue gases of some steam power plants can be utilised in fluorocarbon power plants to give additional power as described in chapter 1 (Part B) of this thesis.

B. Sub-Critical Cycles:

(i) Refrigerant power plants using sub-critical cycles give low efficiencies and hence should not be adopted for generation of power.

C. Brayton Cycles:

A carbon tetrafluoride plant yields an efficiency of 31.8 percent, which is higher than super critical cycles. Such plants can, therefore, be adopted for generation of power.

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