PRINTED IN U.S.A.

69–GT–102

an ASME publication

$1.50 per copy – $.75 to ASME Members

REAL GAS EFFECTS IN CARBON DIOXIDE CYCLES

G. ANGELINO

Lecturer of Special Power Plants
Politecnico, Milan, Italy

The potential performance of carbon dioxide as working fluid is recognized to be similar to that of steam, which justifies thorough thermodynamic analysis of possible cycles.

The substantially better results achievable with CO^2 with respect to other gases are due to the real gas behaviour in the vicinity of the Andrews curve. Simple cycles benefit from the reduced compression work, but their efficiency is compromised by significant losses caused by irreversible heat transfer. Their economy, however, is appreciably better than that of perfect gas cycles. More complex cycle arrangements, six of which are proposed and analyzed in detail, reduce heat transfer losses while maintaining the advantage of low compression work and raise cycle efficiency to values attained only by the best steam practice.

Some of the cycles presented were conceived to give a good efficiency at moderate pressure which is of particular value in direct-cycle nuclear applications.

The favourable influence on heat transfer coefficients of the combined variation with pressure of mechanical, thermal and transport properties, due to real gas effects, is illustrated. Technical aspects as turbo-machines dimensions and heat transfer surfaces needed for regeneration are also considered.

Cooling water requirements are found to be not much more stringent than in steam stations.

Contributed by the Gas Turbine Division of The American Society of Mechanical Engineers for presentation at the International Gas Turbine Conference & Products Show, Cleveland, Ohio – March 10-13, 1969.

NOMENCLATURE

A, B, C, D, E, F, G, H and I	cycle denominations, Figs. 1 and 2
c_p	specific heat at constant pressure, kcal/kg-deg C
h	entalpy kcal/kg
h/h^o	ratio of real gas to perfect gas heat transfer coefficient at a given temperature and pressure
k/k^o	ratio of real gas to perfect gas heat conductivity at a given temperature and pressure
L_u	useful or net work, kcal/kg
p	pressure, atm
p_1	compressor or pump inlet pressure, atm
p_3	turbine inlet pressure, atm
Q_1	primary heat, kcal/kg
Q_2	waste heat, kcal/kg
s	entropy, kcal/kg-degC
$S^* = \Delta h_r / L_u \Lambda$,	heat transfer parameter, proportional to heat exchange surface per unit power, $degC^{-1}$
t	temperature, deg C
t_6	heater inlet temperature, deg C
T	absolute temperature, deg K
U	overall heat transfer coefficient, kW/m^2-deg C
α	condensed fraction of working fluid
ε	expansion ratio
$\Delta\eta$	loss from Carnot cycle efficiency
Δh_r	regenerated heat, kcal/kg
Δs	entropy production, kcal/kg-deg C
Δt_1, Δt_2 and Δt_3	terminal temperature differences of regenerators
η	efficiency
Λ	log mean temperature difference
μ/μ^o	ratio of real gas to perfect gas viscosity at a given temperature and pressure
ρ/ρ^o	ratio of real gas to perfect gas density at a given temperature an pressure

Subscripts

1-11, 9^* and 10^*	define points of cycles, Figs. 1 and 2
c	compressor or Carnot cycle
cr	critical
max	maximum
min	minimum
Q_1	primary heat

Superscripts

o	perfect gas.

INTRODUCTION

Typical power cycles are often defined without making explicit reference to any particular working medium. When a specific fluid is selected however, typical configurations are often abandoned in favour of a closer adjustement of the cycle thermodynamics to the peculiar characteristics of the working substance. The Rankine cycle for instance was so profoundly adapted to water that its modern version employing hypercritical pressure, regeneration and reheat cannot even retain its original denomination and is more properly referred to as steam cycle. In other words the nature of the working fluid suggests cycle configurations which are not necessarily found among the classical solutions. From this point of view the position of carbon dioxide is similar to that of water in that the performance of CO_2 cycles cannot be inferred from general thermodynamic considerations, but must be the object of an exclusive analysis.

In addition, carbon dioxide deserves particular attention since it is abundant and inexpensive and enjoies many attracting characteristics. It exhibits a very good thermal stability [1] [1] (up to 1500 deg C and for pressures of interest the decomposed fraction is negligible), a high chemical inertness (considerably better than that of air or steam) and a precious nuclear behaviour (exceedingly small neutron absorption cross section). Besides, due to its extensive use in gas cooled reactors, the technology related to handling large CO_2 quantities in closed circuits is well established.

Furthermore, as it will result from the following study, carbon dioxide shows potential advantages over steam at moderately-high and high temperatures and its utilization as working fluid could make economically valid the direct cycle gas cooled reactor at temperatures as low as 650 or 700 deg C.

Since, in the author's opinion, the field of application of CO_2 overlaps that of steam for large capacity power stations, particular attention will be given to the comparison of the merits of CO_2 and of steam cycles.

The reasons explaining how it is possible to obtain from a cycle which extends basically in the gas state a performance similar to that of steam cycles lie in the real gas effects [°] arising in a region surrounding the critical point. This region, which is inaccessible for other common gases, can be approached in carbon dioxide cycles due to the comparatively high critical temperature of CO_2 (31 deg C).

Real gas behaviour manifests itself through a mechanical effect (i.e. reduction od specific volume and therefore of compression work) and through a thermal effect (i.e. increase of the specific heat of the compressed fluid, leading indirectly to larger irreversibilities in the regenerative transfer of heat). In order to attain the highest possible efficiency full advantage must be taken of the reduction of specific volume and, at the same time, the detrimental effect of the differing heat capacity between the expanded and the compressed fluid must be minimized through a convenient cycle organization. In carbon dioxide as in steam cycles simplicity is abandoned in view of better efficiencies; the best performance being attained from the most complicated cycles.

The aim of this paper is to present the results of the calculated performance of CO_2 cycles of various complexity, from simple Brayton cycles with weak real gas effects to partial condensation, reheat cycles with multiple regeneration.

Some results relating to the same subject with indications about the thermodynamic analysis were presented in [2] (simple condensation cycles), [3] (partial condensation cycles) and [4] (influence of the working fluid nature on the performance of simple cycles) and are not repeated here in detail.

CYCLE CONFIGURATIONS

Cycles A, B, C, D, E, F and G, whose performance is more or less deeply influenced by real gas effects are presented in Fig. 1a to f. Cycle A is a total condensation cycle; cycle C is a subcritical ($p_1 < p_{cr}$) Brayton cycle in which the compression is affected by the non-ideal behaviour of the working fluid; cycle B is the hypercritical ($p_1 > p_{cr}$) Feher's cycle [5] (cycles A, B and C are denominated simple cycles and can be performed according to the flow diagram of Fig. 1a). Cycle D is a partial condensation cycle in which the compression is performed partly in the liquid and partly in the gas phase and which exhibits reduced internal heat transfer irreversibilities [3] ; similarly cycle G can be considered as a modification of cycle B capable of reducing heat transfer losses.

1 Numbers in brackets designate References at end of paper

(°) throughout this paper the term "real gas effects" is used to indicate the departure of the volumetric behaviour of the fluid from the perfect gas law, due to inter molecular actions in the vicinity of the Andrews curve. High temperature phenomena, such as dissociation giving rise to another class of non-ideal effects are not considered here.

Cycle E is obtained from cycle D by lowering the turbine exhaust pressure p_4 below the condensation value p_1; in a similar way cycle G yields cycle F. Alternatively cycles E and F can be considered as the results of the inter-connection of cycle A with cycle C, and of cycle B with cycle C respectively.

It can be observed that in cycles E and F waste heat $h_{10}-h_1$ being available at sufficiently high temperature, can be partially recovered through a proper reorganization of cycle, as shown in Fig. 2, cycles H and I.

The computation of the performance of cycles A,B and C is straightforward.

For the other cycles it is necessary to establish preliminary which fraction of the working fluid is condensed (or cooled to temperature t_1).

The criterion followed here is the same illustrated in [2]; i.e. the cold and hot end temperature differences of the low temperature regenerator Δt_1 and Δt_2 are selected on the base of heat transfer considerations ($\Delta t_1 = t_7 - t_2$ for cycles D and G; $\Delta t_1 = t_9 - t_2$ for cycles E and F; $\Delta t_2 = t_5 - t_8$, for cycles D, G a F). Hence fraction α cooled to point 1 is given by the heat balances:

$\alpha(h_8-h_2) = (h_5-h_7)$ for cycles D and G; $\alpha(h_8-h_2) = (h_5-h_9)$ for cycles E and F. Once α is determined all other calculations are easily accomplished by means of a CO_2 Mollier diagram.

In cycles H and I fraction α is obtained in the following way. After terminal temperature differences $\Delta t_3=t_{10}-t_{11}$ and $\Delta t_2=t_5-t_8$ are chosen points 11, 5 and 10^* ($t_{10^*} = t_{10}$) can be determined.

Fraction α is then given by the balance: $\alpha\ (h_8-h_{11}) = (h_5-h_{10^*})$. It is then necessary to ascertain that thermal energy available along the isobars 9-10 and 9^*-10^* is sufficient to heat the compressed liquid from t_1 to t_{11}. This is done in the following way.

Temperatures t_9 and t_{9^*} are determined through the relationships: $(h_{10^*}-h_{9^*}) + \alpha(h_{10}-h_9) + \alpha(h_{10}-h_9) = \alpha(h_{11}-h_2)$ and $t_{9^*} = t_9$. The solution is acceptable if $t_9 > t_2 + \Delta t_1$. (Δt_1 is suggested by heat transfer considerations).

In case $t_9 < t_2 + \Delta t_1$ a different criterion for the determination of α must be employed. In the present work however only results obtained through the first method are presented.

Component efficiencies and other parameters were chosen as reported in Table 1, unless otherwise stated. Some of them, namely t_1 and Δt were then used as variables to put in evidence their influence on cycle performance.

REAL GAS EFFECTS ON BRAYTON CYCLES-PERFORMANCE OF SIMPLE CYCLES B AND C

Cycles B and C of Fig. 1b can be considered as Brayton cycles in which the compression is strongly affected by the particular state of aggregation of the molecules in the vicinity of the Andrews curve.

The different value of the specific volume in the expansion and in the compression phases which in perfect gas cycles is due only to the different levels of temperature at which expansion and compression are performed, in real gas cycles is stressed by the action of inter-molecular forces. This has a positive influence on specific work, and an indirect, detrimental effect on regeneration of waste heat after the expansion. How these two effects combine to give the overall cycle efficiency is seen in Fig. 3a, whose curve were plotted for different base pressure and for a cold end temperature difference of regenerator t_5-t_2 of 30 deg C. However, the same t_5-t_2 = 30 deg C gives rise to widely differing regenerator surface requirements, as shown in Fig. 3 b in which the heat transfer parameter $S^* = \Delta h_r/L_u\Lambda$ was plotted for various base pressure (S^* is proportional to the exchange surface per unit power, provided the overall heat transfer coefficient U is the same for all cycles. Although, generally speaking, the higher the base pressure, the higher the U which is attainable, beyond a certain pressure the benefit represented by the superior heat transfer coefficient is totally cancelled by the increased cost of the unit exchange surface. For this reason S^* can be considered a meaningful heat transfer parameter over a wide range of variation of the base pressure).

The performance of cycles C and B at p_{min} = 40 and p_{min} = 100 atm respectively, is remarkable both with respect to efficiency and to exchange surface. A more detailed report on the overall efficiency of CO_2 cycles at different p_{min}, p_{max}, t_5-t_2 and S^* is given in Fig. 4.

In order to allow the comparison of cycle performance on a more homogeneous base, t_5-t_2 was varied to give a constant S^* = 0.1 for all cycles, (S^* = 0.1 can be considered a typical value for closed Brayton cycles). The results are reported in Fig. 5.

For a turbine inlet temperature of 700 deg C maximum overall efficiency is 0.340 for a perfect gas cycle, 0.366 at p_{min} = 20 atm p_{max} = 90 atm; 0.385 at p_{min} = 40 atm p_{max} = 150 atm and 0.424 at p_{min} = 100 atm p_{max} = 300 atm, which proves the favourable influence of the non-ideal behaviour of the working fluid. Even superior benefits are achievable thanks to real gas effects for turbine and compressor efficiencies lower than assumed on Table 1. For η_t= 0.85 and η_c= 0.80 cycle efficiency is equal to 0.258 in the ideal gas assumption and increases to 0.338 at p_{min} = 40 atm, as shown also on Fig. 5.

Finally, the influence of the exchange surface, S^* and of the cold end temperature difference of regenerator t_5-t_2 on cycle economy is shown in Fig. 6, from which it can be inferred that the higher the base pressure the higher is the sensitivity of overall efficiency to regenerator surface. In particular at p_{min} = 150 atm cycle efficiency reaches 0.46 provided a value of 0.2 in accepted for S^*.

PERFORMANCE OF COMPOUND CYCLES

The economy of cycle C can be greatly improved if a portion of the working fluid is condensed and compressed in the liquid phase. If the temperature of the cooling water is sufficiently cold so that a true isothermal condensation is possible, the interconnection of cycle C with cycle A gives rise to compound cycles D or E (Fig. 1 d and f) whose efficiency is remarkably better than that of both the parent cycles. This is due to the fact that the fraction of working fluid which is condensed represents an additional degree of freedom allowing the mutual reduction of heat transfer irreversibilities in both cycles C and A [3]. Efficiency and specific work of cycles A and D are compared on Fig. 7. Partial condensation of the working fluid reduces both the waste heat Q_2 (which is practically proportional to the condensed fraction) and the specific work (due to the larger work required in the gas phase compression), the net result of both effects being an improvement of overall efficiency. The beneficial influence on efficiency of lowering the condensation temperature and pressure is apparent from Figs. 7 and 8.

If cooling water at temperature sufficiently low to permit an isothermal condensation is not available, the efficiency of cycle C can be considerably improved by its inter-connection with cycle B to give origine to cycles F and G.

As in cycles D and A, also in cycles F and G the compression is partly performed in the liquid phase(°) which reduces compression work and related losses.

Heat transfer irreversibilities are minimized by a proper selection of the condensed fraction. The effect of the addition of a condensation portion to cycle C is illustrated on Fig. 9 giving efficiency and specific work for different maximum and minimum pressures, both for cycle C and for the derived cycle F. The inspection of Fig. 9 suggests the following observations: - the efficiency of cycle F is consistently growing from p_{min} = 10 to p_{min} = 70 atm; similarly the efficiency gain of cycle F over cycle C is growing with p_{min}; at 700 deg C turbine inlet temperature the best efficiency of cycle C is 0.379, whereas the maximum performance of cycles F and G are 0.467 and 0.477 respectively; - specific work of cycle F at maximum cycle efficiency is about twice that of cycle C; this being due partly to the reduced compression work of the liquid phase and partly to

(°) In some cases the working fluid is not, strictly speaking, in the liquid state; what is meant here however is that specific volume and compression work are comparable with values which are typical for the liquid state.

the higher turbine expansion ratios of the compound cycles; - curves giving the performance of cycles F and G are rather "flat,,, which represents a valuable characteristic at partial load operation.
The dependence of overall efficiency of cycles F and G on turbine inlet pressure is particularly evident in low temperature cycles (see Fig. 10). At t_{max} = 400 deg C maximum cycle efficiency is 0.116 for p_{min} = 10 atm, and 0.315 for p_{min} = 75 atm, the latter representing a remarkable performance which is surpassed only by steam cycles. High temperature cycles are less sensitive to p_{min}. At t_{max} = = 800 deg C efficiencies in excess of 0.50 are attained for minimum pressures higher than about 40 atm.
Heat transfer surfaces required for regeneration are completely acceptable as shown in Fig. 11 (for p_3 = 250 atm, p_{min} = 45 atm an exchange surface S^* = 0.08 is sufficient to give 0.47 overall efficiency). It is clear from Fig. 11 that high pressure cycles yieldthe best performance both with respect to efficiency and heat transfer surfaces.
For the lowest values of p_{min} (cycles E or F) temperature t_{10} at the end of the first stage of compression is sufficiently high to permit the renegeration of part of the waste heat $h_{10}-h_1$, with an appreciable increase of cycle economy. In addition, when p_{min} is small high expansion ratios p_{max}/p_{min} are technically acceptable even at moderate p_{max}, in which case reheating becomes particularly advantageous.
The performance of all the cycles so far presented, including the reheat solution is given in Fig. 12 (turbine inlet pressure p_3= 130 atm) and in Fig. 13 (p_3 = 300 atm), as a function of temperature in the interval 500-1000 deg C.
The efficiencies of a double reheat steam cycle [6] and of a perfect gas cycle with inter-refrigerated compression are also reported.
The inspection of Figs. 12 and 13 suggests the following conclusions.
At moderate p_3 (Fig. 12) the efficiency of cycles A and G is very poor and inferior to that of the perfect gas cycle. On the contrary cycles H and I, in particular if one stage of reheating is added, yield a remarkable performance. Cycle efficiency of 0.50 is attained by the reheat H cycle at t_{max} = 640 deg C by the reheat I cycle at t_{max} = 730 deg C, by the double reheat steam cycle at t_{max} = 760 deg C and by the perfect gas cycle at t_{max} = 950 deg C.
At higher p_3 (Fig. 13) all CO_2 cycles are appreciably superior to the perfect gas configuration.
Cycles D and H with one stage of reheating yield better performance than the double reheat steam cycle at temperatures higher than 620 deg C. If the minimum temperature of the working fluid cannot be lowered below 30 deg C (cycles G and I) the limit of superiority of CO_2 cycles is shifted to 800 deg C.
The efficiency curve of cycle A lies between those of steam and perfect gas cycles within the interval 600-1000 deg C; simplicity, prime mover compactness and reduced heat transfer surface requirements can make it nevertheless attractive.
The comparison of Figs. 12 and 13 with Fig. 5 put in evidence the much higher economy of compound over simple cycles.

DISTRIBUTION OF LOSSES

It is known that irreversible processes taking place in a typical component account for an efficiency loss with respect to that of Carnot cycle, given by:
$\Delta\eta = T_{min}\Delta s/Q_1$, in which Δs is the entropy production within said component. Accordingly, cycle efficiency is given by: $\eta = \eta_C - \Sigma\Delta\eta$.
When computing the various Δs the entropy variation of the hot and of the cold source exchanging heat at t_{max} and t_{min} respectively must be taken into account.
Simple cycles were analyzed on this base in [4], compound cycles at a higher efficiency level are considered here (Fig. 14). Maximum and minimum temperatures being 700 and 30 deg C respectively for all cycles, the Carnot cycle efficiency to which losses must be subtracted is equal to 0.689. In perfect gas cycles with inter-refrigerated compression (Fig. 14a), fluid-dynamic losses (compressor, turbine, pressure drops) although inferior to heat transfer inefficiencies represent a substantial fraction of the overall loss (about 40%). At the opposite end, in Feher's cycle inefficiencies due to heat transfer (of waste heat Q_2, primary heat Q_1 and regenerated heat, Fig. 14b) account for more than 85% of the total loss. Regenerator loss by itself is higher than the sum of all other losses.
The compound configuration (cycle G, Fig. 14c) reduces substantially regenerator losses at the expenses of increased mechanical (i.e. fluid-dynamic) dissipations which represent 30 to 40% of the total loss.
In cycle F the transfer of primary and of waste heat departs more significantly than in cycle G from an isothermal process which justifies the increased importance of related losses (Fig. 14d).
A better but more complicated organization of cycle F which gives rise to cycle I reduces the inefficiency connected with the transfer of waste heat to the cold source (Fig. 14e).
Loss $\Delta\eta_{Q1}$ due to the transmission of primary heat to the working fluid is reduced or even cancelled if heat is available from the hot source at a decreasing temperature, in which case the hot source itself is partly responsible of the efficiency drop with respect to η_C. A similar situation is encountered in gas cooled nuclear reactors and in waste heat recovery systems. In other words there are cases in which loss $\Delta\eta_{Q1}$ cannot be ascribed to the power cycle.
When this happens a better measure of intrinsic cycle inefficiency is represented by $\Sigma\Delta\eta - \Delta\eta_{Q1}$ instead that by the simple $\Sigma\Delta\eta$. Under this respect at each turbine inlet pressure there is a cycle pressure level (p_{min}) which minimizes $\Sigma\Delta\eta - \Delta\eta_{Q1}$, as shown in Fig. 15. The lower the turbine inlet pressure, the lower the optimum cycle pressure level.
It is known that $\Delta\eta_{Q1}$ is by far the most important loss in steam cycles, in which other inefficiencies, on the contrary, are very limited. A number of Rankine cycles are compared on this base with two carbon dioxide cycles at p_{min} = 40 and p_{min} = 75 atm (cycles F and G respectively) on Fig. 16. The extremely low compression work, the good efficiency of modern turbines and the highly effective regenerative heating of water of present-day stations account for a remarkably small $\Sigma\Delta\eta - \Delta\eta_{Q1}$.
On the contrary, the nature of the working medium and, in particular, the shape of the Andrews curve are responsible for the very high $\Delta\eta_{Q1}$, particularly at elevated turbine inlet temperatures.
In CO_2 cycles losses are distributed much more uniformly.

TECHNICAL ASPECTS

It was shown in the previous sections that compound CO_2 cycles can compete, with respect to efficiency, with steam cycles over a wide range of pressures and temperatures. However technical problems will have a decisive influence on the final economy of power produced by means of CO_2 cycles.
The greatest benefits of the utilization of CO_2 in place of steam relate to the prime mover. Carbon dioxide expansion work is 5 to 10 times smaller than the specific work of a modern steam cycle (Fig. 17).
Consequently the number of stages of the CO_2 prime mover will be only a small fraction of that of steam turbines. Radial dimensions of turbomachines are essentially dependent on volume flow rates. Exhaust volume flow is particularly important in determining the maximum turbine radial dimensions. As shown in Fig. 18 CO_2 exhaust flow per unit power is 30 to 150 times less than that of steam.
On the same figure it can be seen that,in steam cycles, volume flow at the turbine admission after the second reheat of steam almost equals that of the exhaust of high pressure CO_2 cycles.
As a consequence the radial dimension of CO_2 turbines can be extremely small even for the greatest output (Fig.18c).
If turbine inlet temperatures higher than 650 deg C are to be employed, rotating blades cooling becomes compulsory.

In a double reheat steam cycle more than 150 kcal/kg must be converted into work in order to reduce the steam temperature from 750 to 650 deg C at the beginning of the expansion and after each reheating. Consequently the number of stages to be cooled is very large and the complexity of the prime mover prohibitive. On the contrary, as shown on Fig. 19, in CO_2 cycles an entalpy drop of 29 kcal/kg (converted into work by 2 or 3 stages) is sufficient to cool the working fluid from 750 to 650 deg C. From this point of view too for high temperature operation carbon dioxide seems preferable to steam.

For all power cycles it is of interest to know the combined capacity of turbomachines which is required to produce the unit power. Various CO_2 cycles are compared on this base on Fig. 20.

In Brayton cycles C the combined power of turbomachines is 2 to 4 times the net output, depending on the intensity of the real gas effects (the highest value relates to perfect gas cycles).

In other simple and compound cycles the total power of turbine, pump and compressors is 1.2 to 1.8 times the generator capacity.

A significant parameter in the design of primary heater and regenerator is represented by the heater inlet (or regenerator outlet) temperature t_6.

The cost of the unit exchange surface of both external heater and regenerator is increasing with t_6. Simple and compound CO_2 cycles are analyzed with reference to this aspect on Fig. 21.

Compound cycles are characterized by more favourable heater inlet temperatures than simple cycles. A remarkable exception is represented by cycle A which exhibits the lowest t_6 and a reasonably good efficiency.

Of particular value is also the performance of cycle H.

The favourable influence of low condensation temperatures on both efficiency and t_6 is also shown on Fig. 21 with reference to cycles D and G.

The possibility of low temperature condensation lies in the availability of abundant and cold cooling water. The amount of cooling water required for condensation at different temperatures (cycles D and G) is given on Fig. 22.

The operating points of a number of steam condensers are also reported. From the inspection of Fig. 22 it can be inferred that heat is rejected to the ambient from CO_2 cycles at conditions which are not appreciably worse than those encountered in typical steam practice. Accepting the current value of 100 tons of cooling water per hour and per Mw, condensation can be performed at 15 or 23 deg C provided the cooling water temperature is 5 or 15 deg C respectively.

REAL GAS EFFECTS ON HEAT TRANSFER COEFFICIENTS

The influence of real gas effects on cycle efficiency is chiefly due to the reduction of specific volume and compression work in the vicinity of the Andrews curve and to the variation of heat capacities not only with temperature but also with pressure.

In addition the real gas behaviour has a remarkable influence on transport properties and on heat transfer coefficients. Assuming that also in hypercritical regions typical correlations of turbulent flow [7] mantain their validity, the following expression can be derived for the ratio of real gas to perfect gas heat transfer coefficients at a given pressure and temperature along a tube having a fixed diameter and length, when the frictional pressure loss of both real and perfect gas flows are the same (°):

$$h/h^o = (\rho/\rho^o)^{0.445}\,(c_p/c_p^{\,o})^{0.4}\,(k/k^o)^{0.6}\,/(\mu/\mu^o)^{0.489}$$

According to this expression heat transfer is affected by the correlated variation of mechanical thermal and transport properties. A more detailed, quantitative analysis shows that, at pressures of interest, all these parameters have a positive influence, i.e.: $\rho/\rho^o > 1$; $c_p/c_p^{\,o} > 1$ and $(k/k^o)^{0.6}/(\mu/\mu^o)^{0.489} > 1$. Their combined effect is reported on Fig. 23, from which it is clear that definite benefits in the transfer of heat are to be expected even on the low pressure side (50-60 atm) of the regenerator where the heat transfer requirements are more stringent. The favourable influence of real gas effects on heat exchange is appreciably larger if it is taken into account the fact that, due to the higher heat transfer coefficients, shorter tubes are needed to exchange a given amount of heat.

CONCLUSIONS

From the results reported in the previous sections it can be inferred that real gas effects if properly put at work, represent a powerful tool to raise cycle economy. For a cooling water temperature of 5 deg C available in a quantity similar to that required by steam plants, at a turbine inlet temperature of 700 deg C cycle efficiencies better than that of a double reheat steam cycle at the same maximum temperature and in excess of 0.50 are achievable.

The superiority of reheat CO_2 cycles over double reheat steam cycles at t_{max} = 700 deg C is mantained up to a cooling water temperature of about 20 deg C.

Other advantages over steam cycles are the simpler layout, and the smaller and simpler prime mover. Furthermore CO_2 cycles will benefit much more appreciably from any technological improvement allowing the use of higher turbine inlet temperatures. With respect to perfect gas cycles, at t_{max} = 700 deg C the CO_2, real gas configuration in 25% more efficient and requires less than half the combined power of turbomachines; on the contrary the perfect gas arrangement enjoies a wider freedom with respect to pressure level and minimum temperature selection.

Even at low temperature CO_2 compound cycles have attractive characteristics as shown by the fact that a cycle efficiency of 0.38 is attainable through non-reheat cycles at 500 deg C turbine inlet temperature, and 20 deg C cooling water temperature (η = 0.44 for a reheat steam cycle at the same t_{max}).

In many of the cycles considered an important fraction of losses is due to irreversible heat transfer in the regenerator.

The analysis of heat exchange deserves therefore a considerable attention. Preliminary considerations show that real gas effects have an important and favourable influence also on heat transfer coefficients.

(°) It is known that in the hypercritical state usual heat transfer correlations must be somewhat modified. If heat is exchanged through small temperature differences, however, as in power cycle regenerators and in regions where physical properties do not vary abruptly the ordinary heat transfer theory is probably still adequate.

REFERENCES

1 Bailey, H.E., "Equilibrium thermodynamic properties of carbon dioxide", NASA SP-3014, Washington, D.C. 1965.

2 Angelino, G., "Perspectives for the liquid phase compression gas turbine", Journal of Engineering for Power, Trans. ASME, Series A, vol. 89, n.2, Apr. 1967, pagg. 229-237.

3 Angelino, G., "Carbon dioxide condensation cycles for power production", ASME paper n. 68-GT-23 presented in Washington, D.C., March 1968.

4 Casci, C. and Angelino, G., "The dependence of power cycle performance on their location relative to the Andrews curve", ASME paper to be presented in Cleveland, March 1969.

5 Feher, E.G., "The supercritical thermodynamic power cycle", in Advances in Energy Conversion, The American Society of Mechanical Engineers, New York, N.Y., 1967, pp. 37-44.

6 Rydgkin, V.I. and Tambieva, I.N., "Perspectives for the improvement of efficiency of large thermal power stations", Teploenergetika, n. 11, November 1960, pp.9-15.

7 McAdams, W.H., Heat Transmission, McGraw-Hill Book Co.,Inc., New York, N.Y., and Kogakusha Co., Tokyo, third edition, 1954, p. 219.

TABLE 1

CHARACTERISTIC PARAMETERS FOR CARBON DIOXIDE CYCLES

Parameters	Value
Turbine adiabatic efficiency, η_t	0.90
Compressor adiabatic efficiency, η_c	0.85
Pump adiabatic efficiency, η_p	0.85
Cycle fractional pressure drop, $\Sigma\Delta p/p$	0.15
Minimum temperature of cycles A, D, E and H t_1	15 deg C
Minimum temperature of cycles B, C, G, F and I t_1	30 deg C
Cold end temperature difference of regenerator, cycles B and C, (t_5-t_2)	30 deg C
Cold end temperature difference of low temperature regenerator, cycles D and G, (t_7-t_2)	15 deg C
Cold end temperature difference of low temperature regenerator, cycles E and F, (t_9-t_2)	15 deg C
Cold end temperature difference of high temperature regenerator, cycles D, G, E, F, H and I, (t_5-t_8)	30 deg C
Hot end temperature difference of low temperature regenerator, cycles H and I, $(t_{10}-t_{11})$	15 deg C

CAPTIONS OF FIGURES

1 Different cycle configurations employing carbon dioxide as working medium and profiting by real gas effects.

2 Compound cycles yielding high efficiency at moderate turbine inlet pressures.

3 Efficiencies and heat transfer surface requirements for simple cycles B and C.

4 Performance of simple cycles as a function of base pressure and regenerator temperature difference; Δt_1 is the cold end temperature difference of regenerator.

5 Comparative performance of simple cycles B and C at a given S and influence of the efficiency of turbomachines on perfect and real gas cycles.

6 Sensitivity of the efficiency of cycles B and C to heat transfer surface and to cold end temperature difference of regenerator.

7 Comparative performance of cycles A and D and influence of the condensation temperature on cycle D.

8 Influence of the condensation temperature on various characteristic parameters of cycle D.

9 Influence of the addition of a gas phase compression to cycle C in order to reduce heat transfer irreversibilities.

10 Performance of cycles F and G for base pressures ranging between 10 and 75 atm.

11 Heat transfer surface required for regeneration in cycles F and G as a function of cycle base pressure.

12 Summary of the performance of different carbon dioxide cycles as compared with double reheat steam and inter-refrigerated compression perfect gas cycles at a moderate turbine inlet pressure (130 atm).

13 Summary of the performance of different carbon dioxide cycles as compared with double reheat steam and inter-refrigerated compression perfect gas cycles at high turbine inlet pressure (300 atm).

14 Losses analysis for an inter-refrigerated compression perfect gas cycle, for simple cycle B and compound cycles F, G and H.

15 Sum of losses which do not depend upon external heat source for cycles F and G.

16 Comparison of losses of steam and CO_2 cycles.

17 Turbine expansion work in steam and carbon dioxide cycles.

18 Exhaust volume flow rates for carbon dioxide and steam cycles and comparative dimensions of turbine exhaust annuli. Volume flow rates after the second reheat of a double reheat steam cycle are also reported for comparison.

19 Approximate number of stages requiring some sort of cooling for double reheat steam and carbon dioxide cycles at high temperature operation.

20 Combined power of turbomachines (turbine plus compressor and pump) per unit power output for different carbon dioxide cycles.

21 Cycle efficiency as a function of primary heater inlet temperature.

22 Cooling water requirements for condensation of carbon dioxide at different temperatures compared with steam condensers current practice.

23 Real gas effects on turbulent flow heat transfer coefficients.

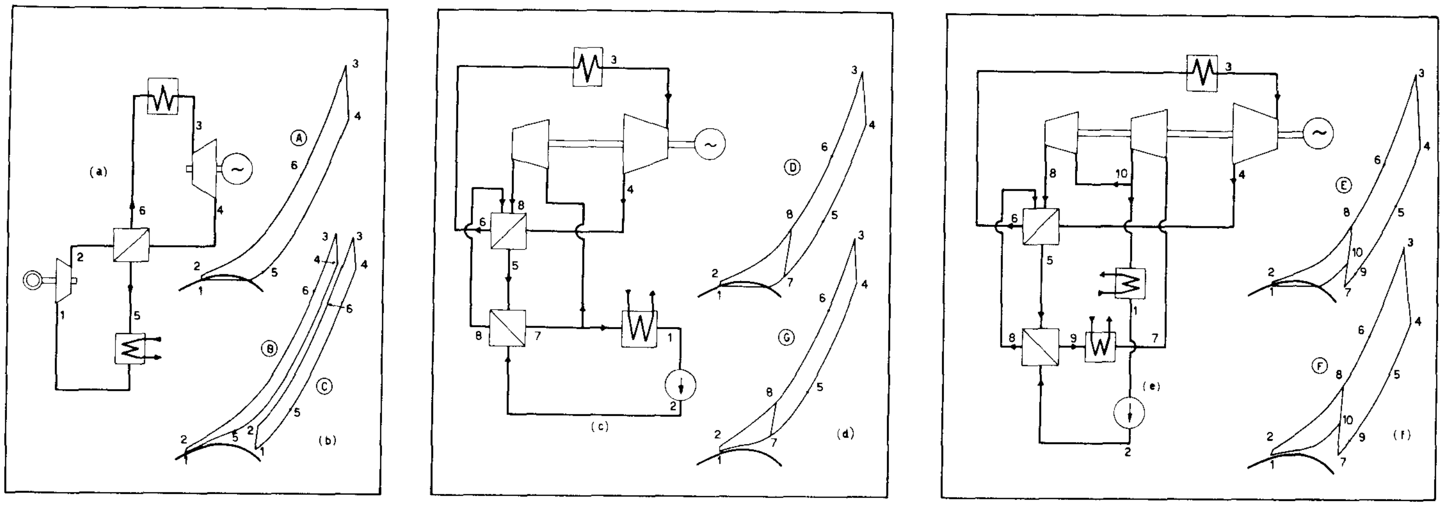


Figure 1

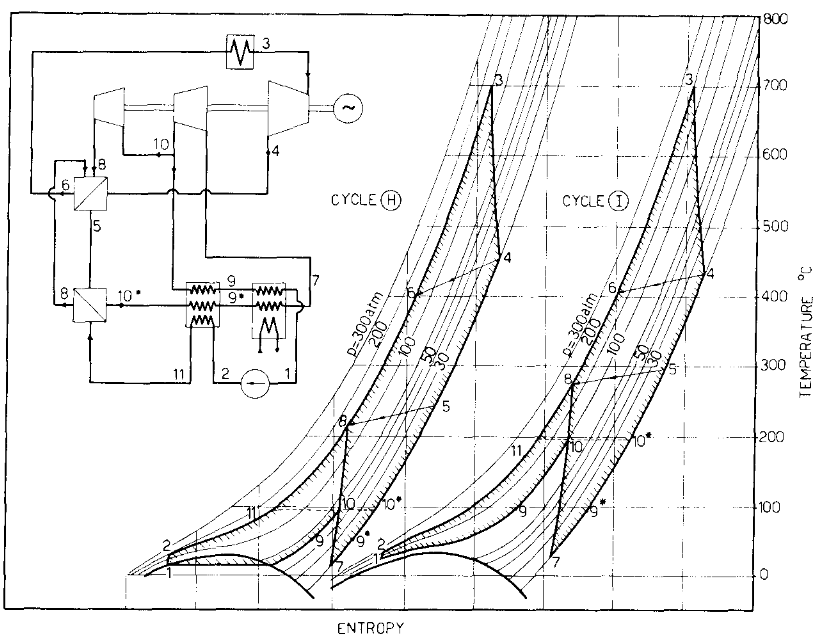


Figure 2

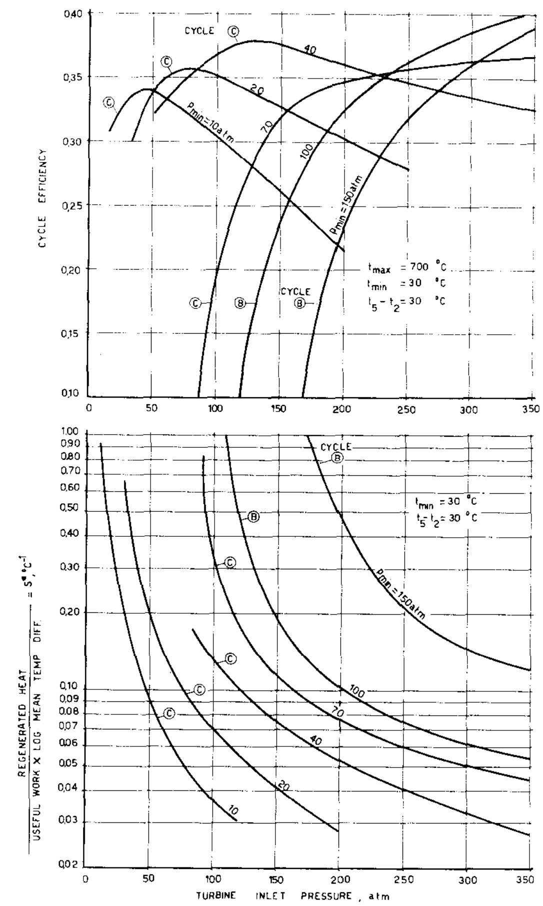


Figure 3

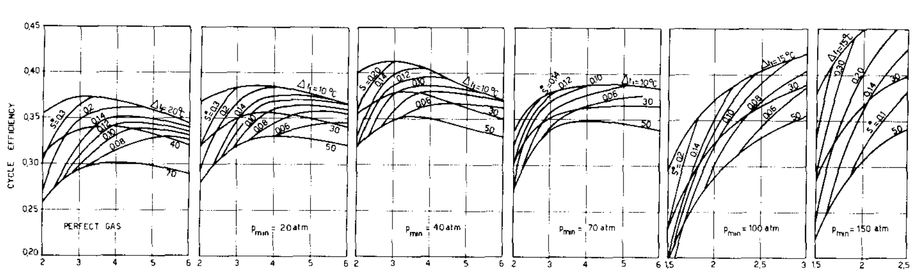


Figure 4

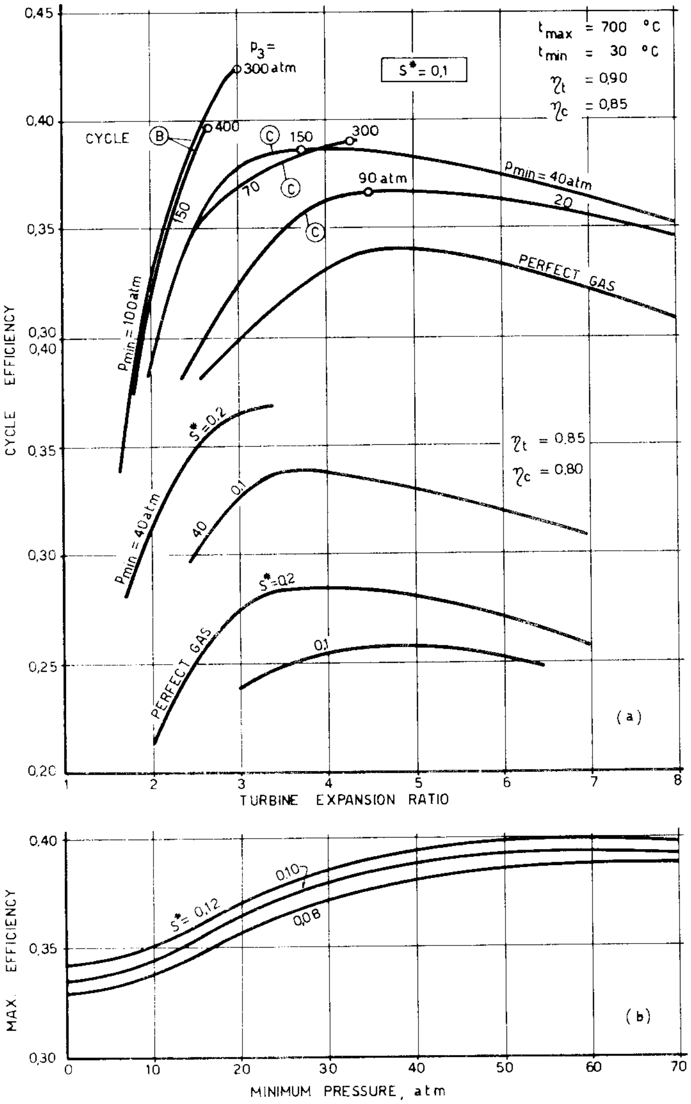


Figure 5

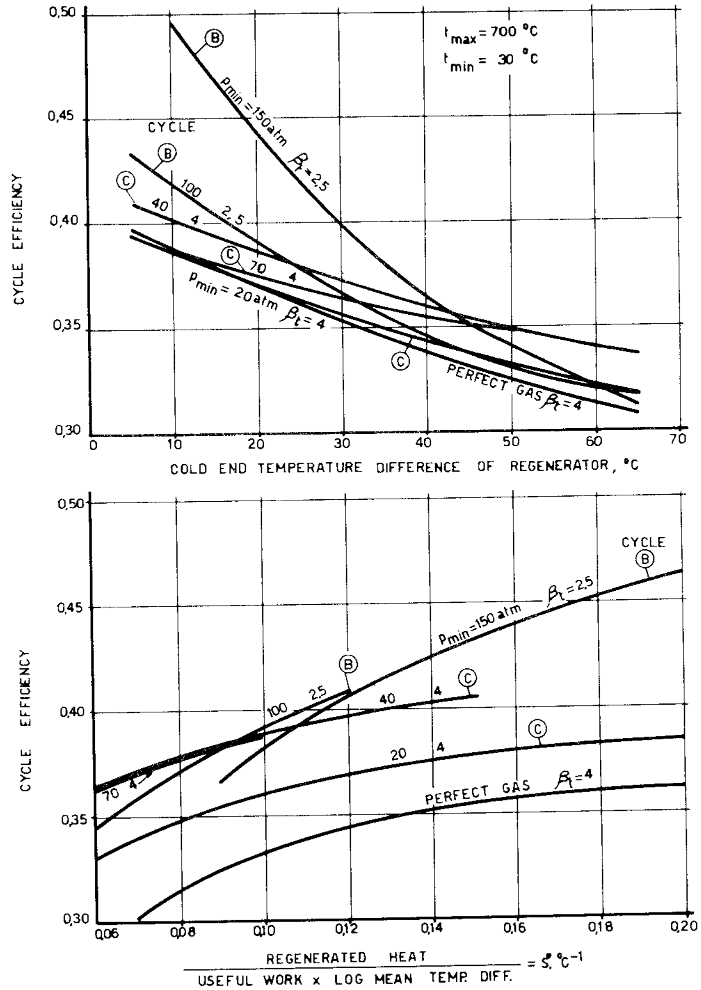


Figure 6

Figure 7

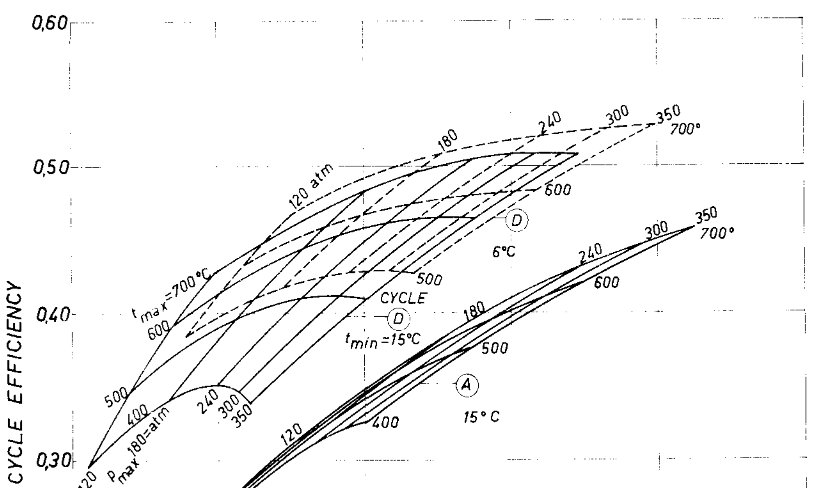
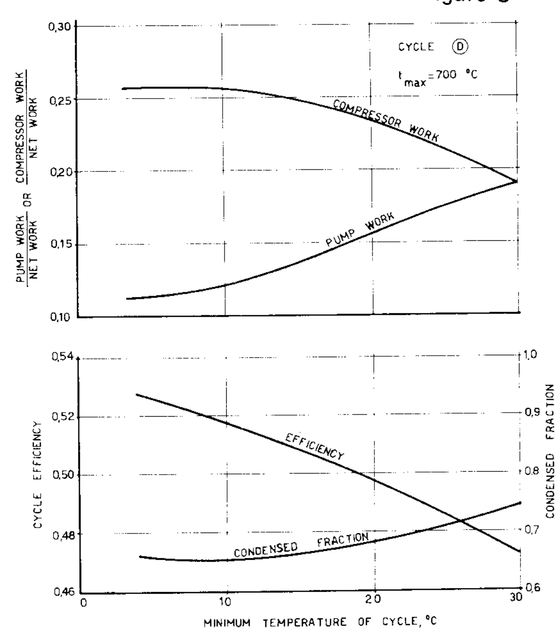


Figure 8



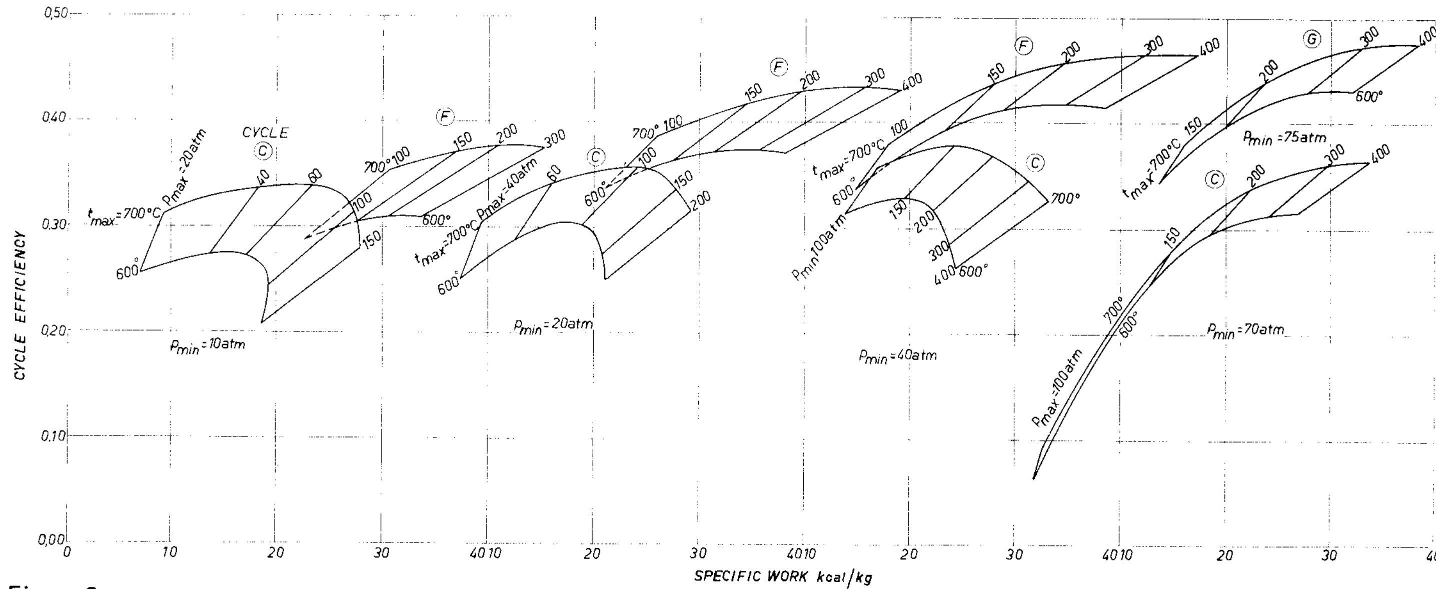
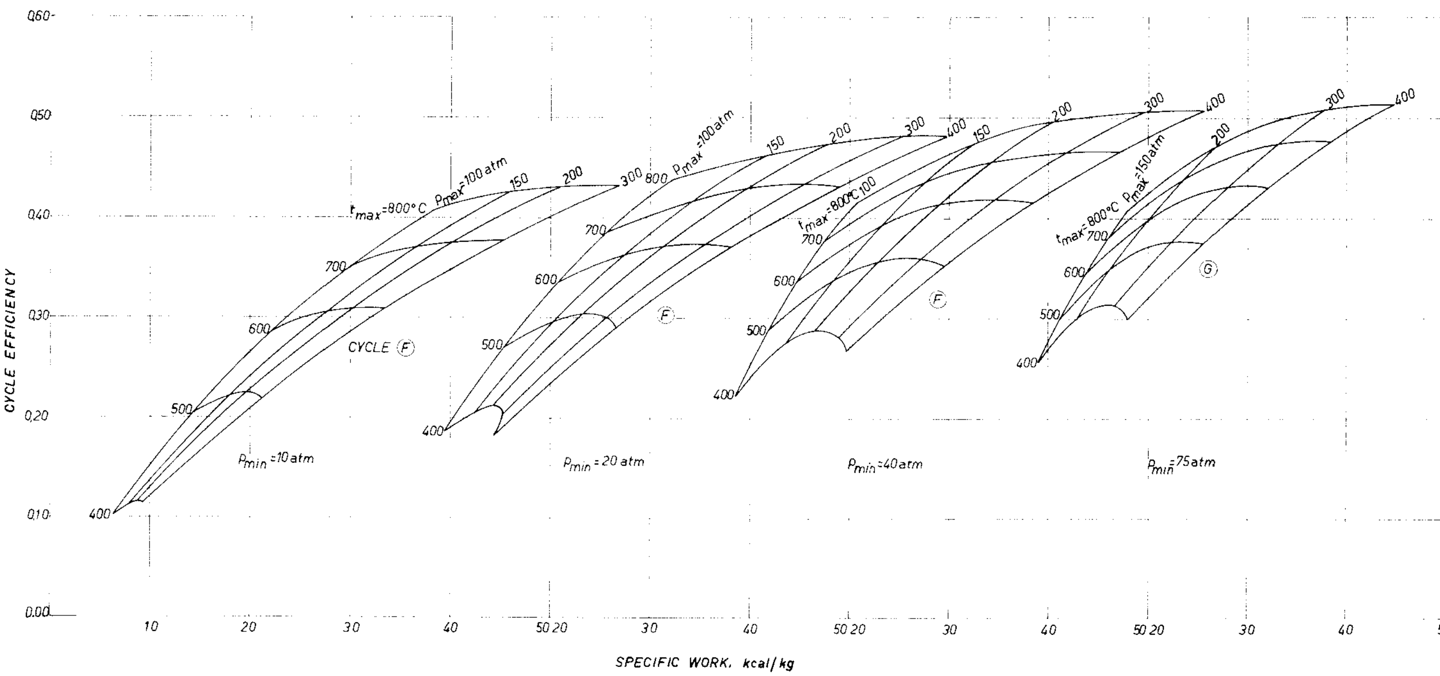


Figure 9

Figure 10



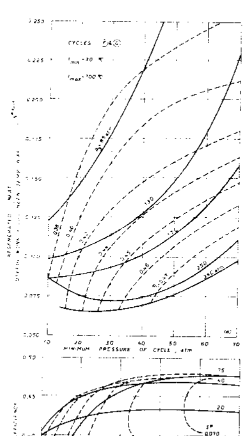


Figure 11

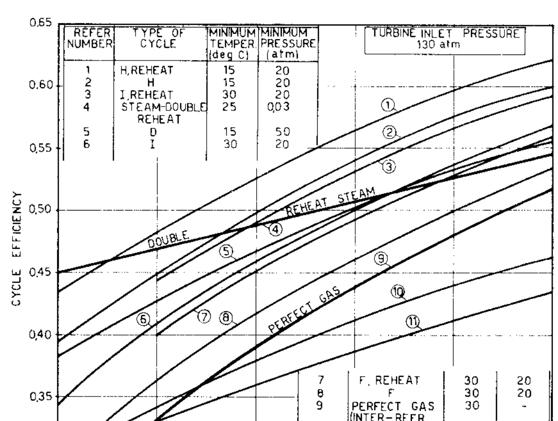


Figure 12

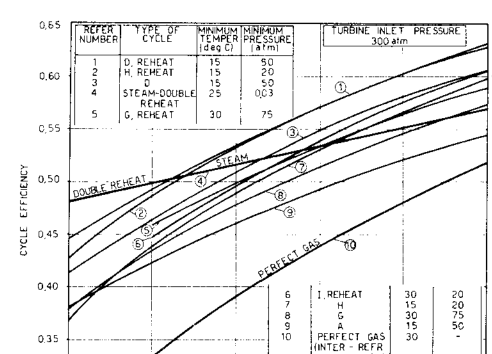


Figure 13

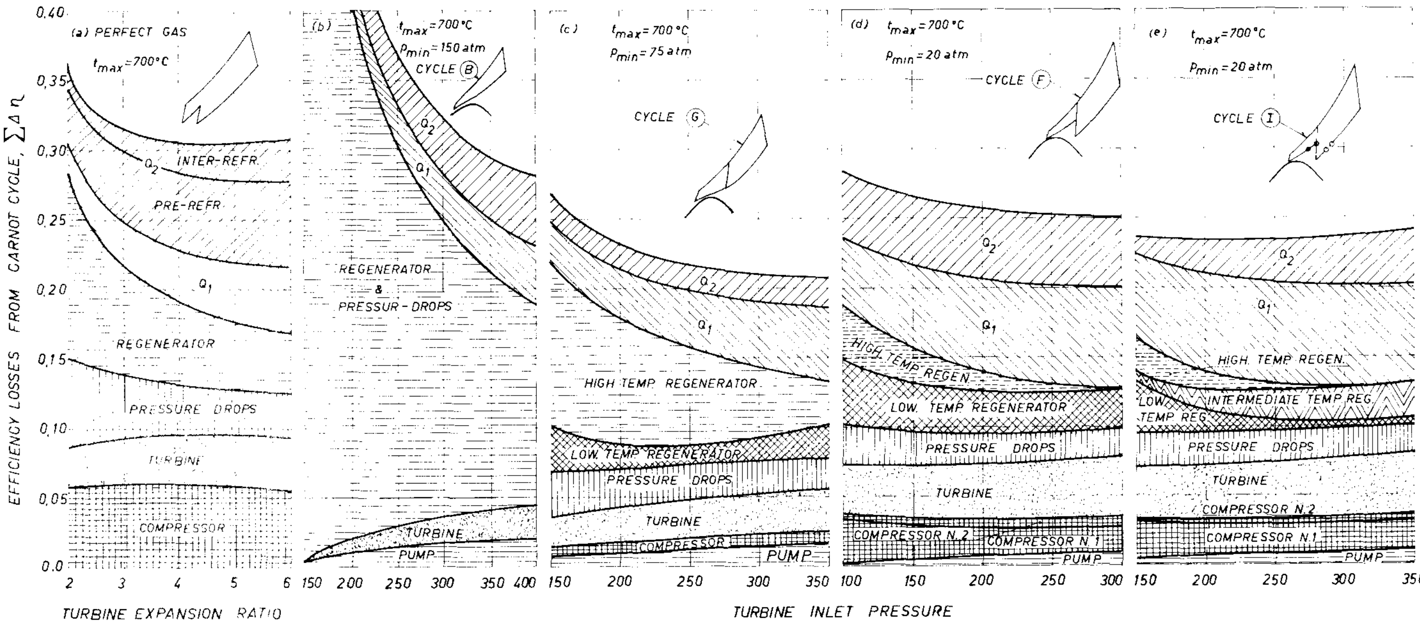


Figure 14

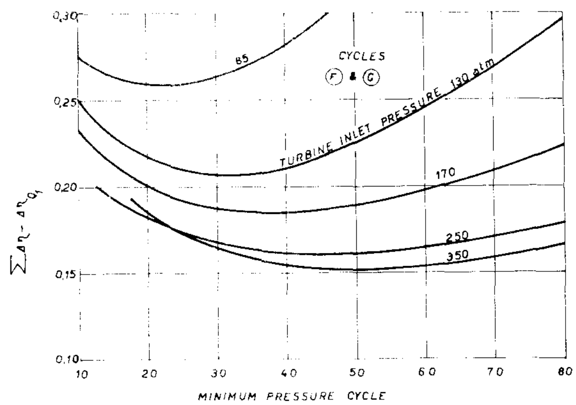


Figure 15

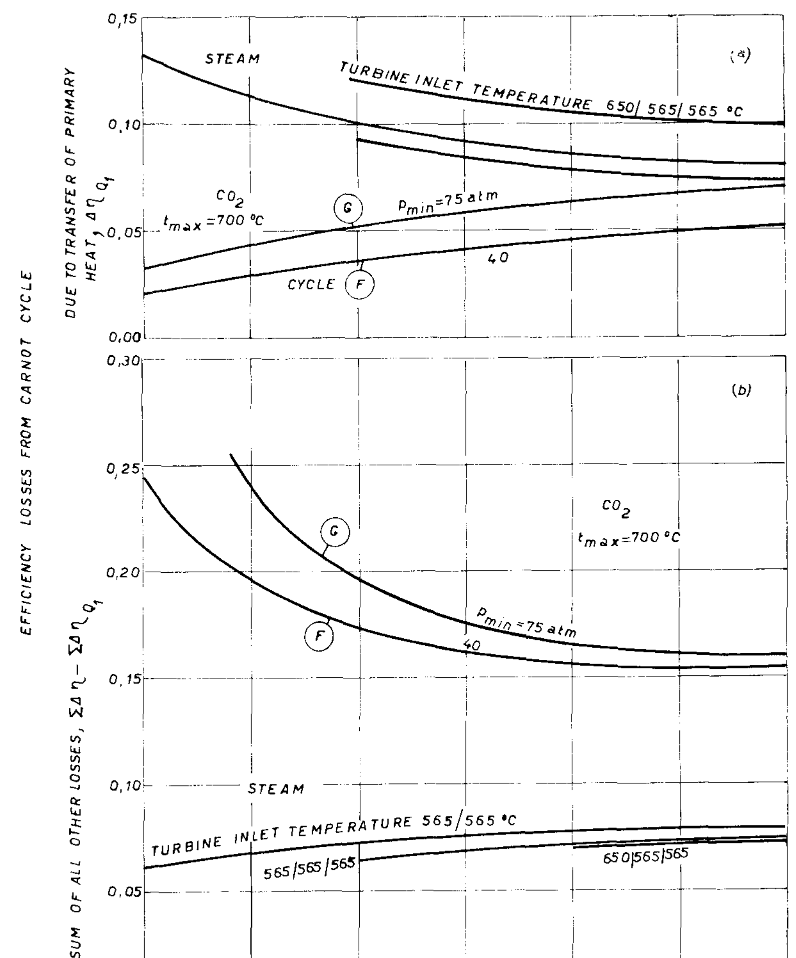


Figure 16

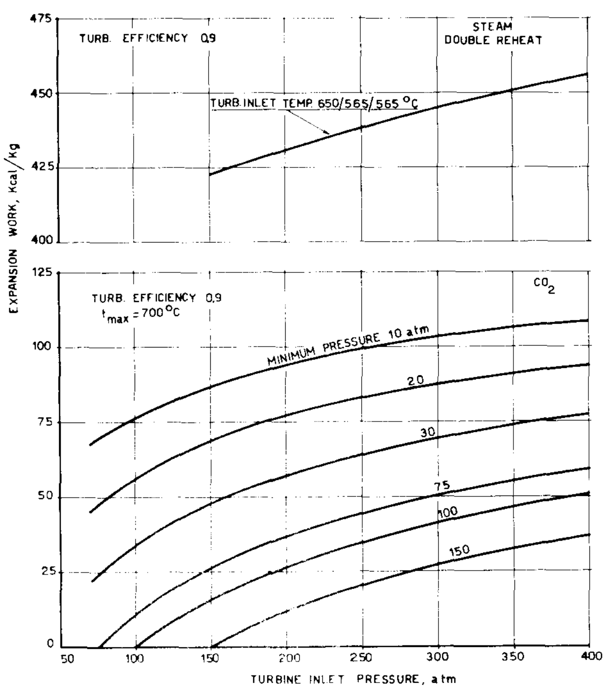


Figure 17

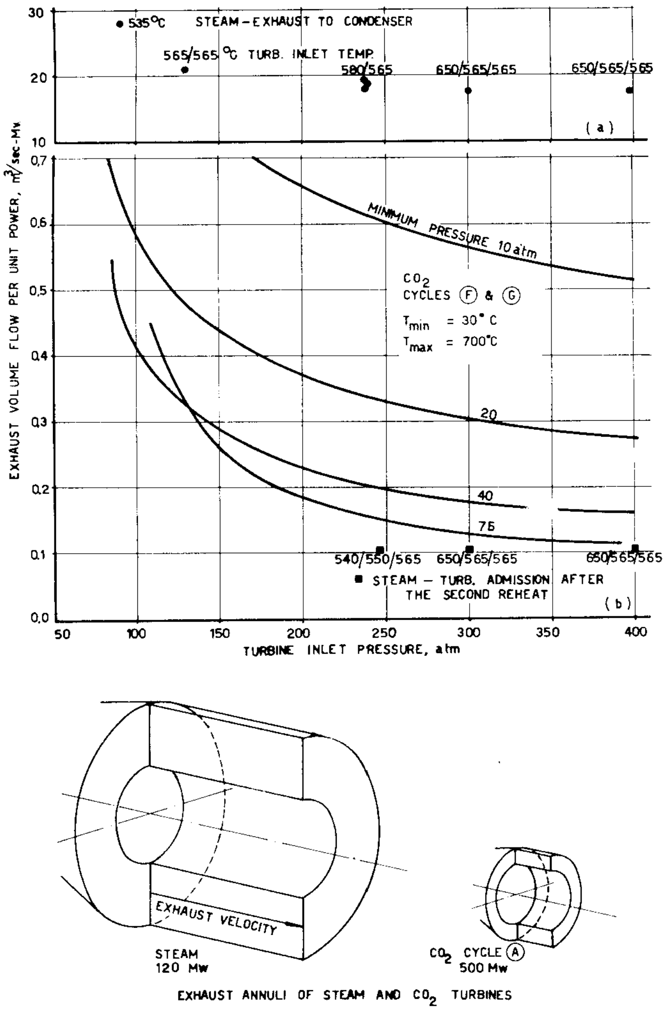


Figure 18

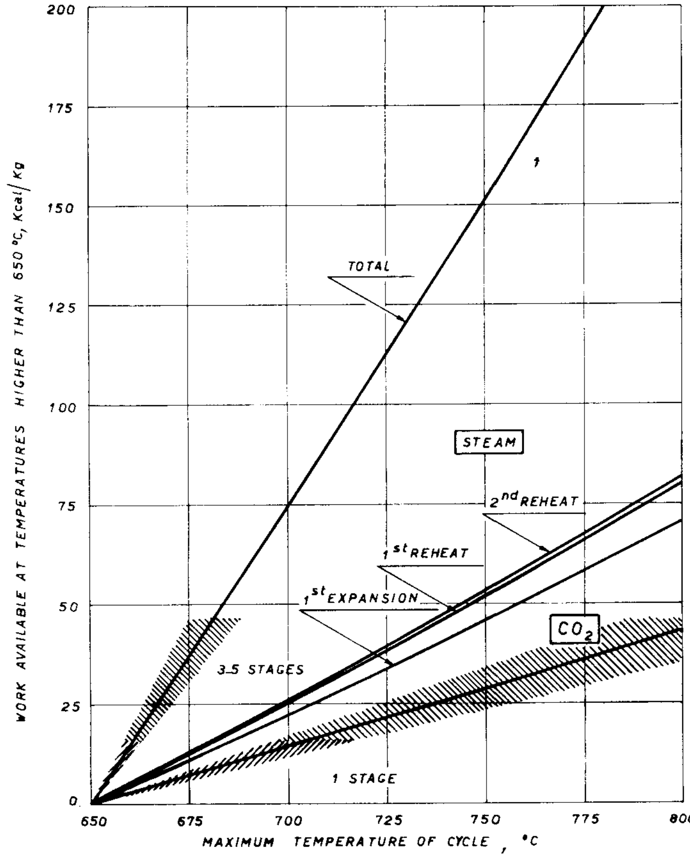


Figure 19

Figure 20

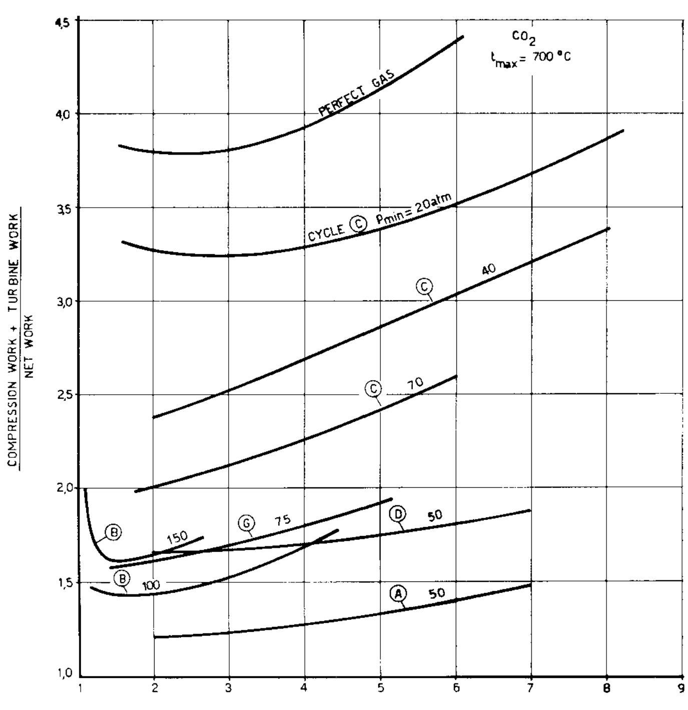
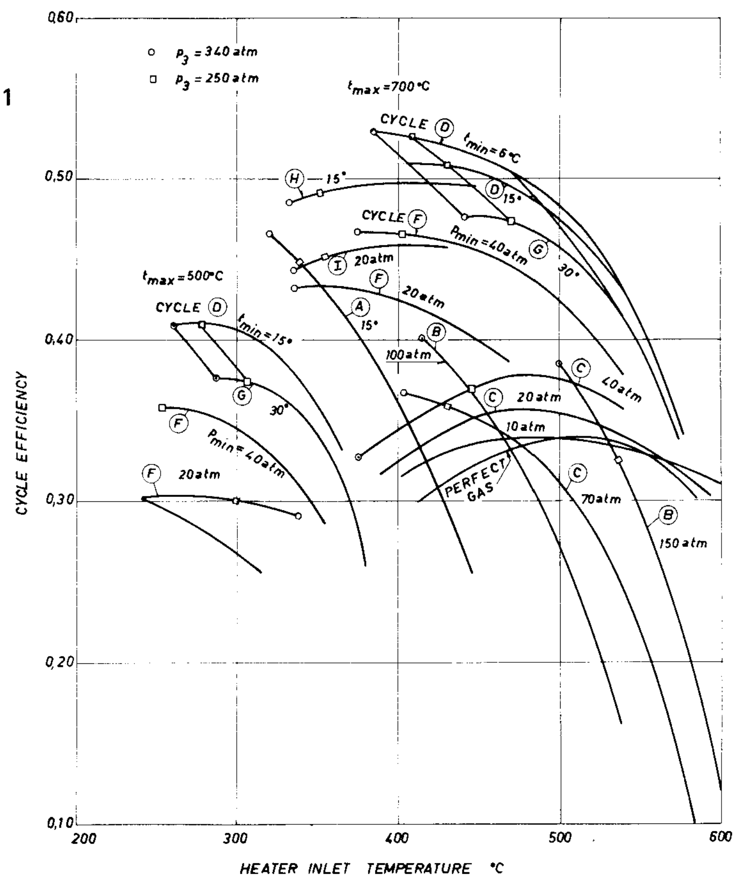


Figure 21



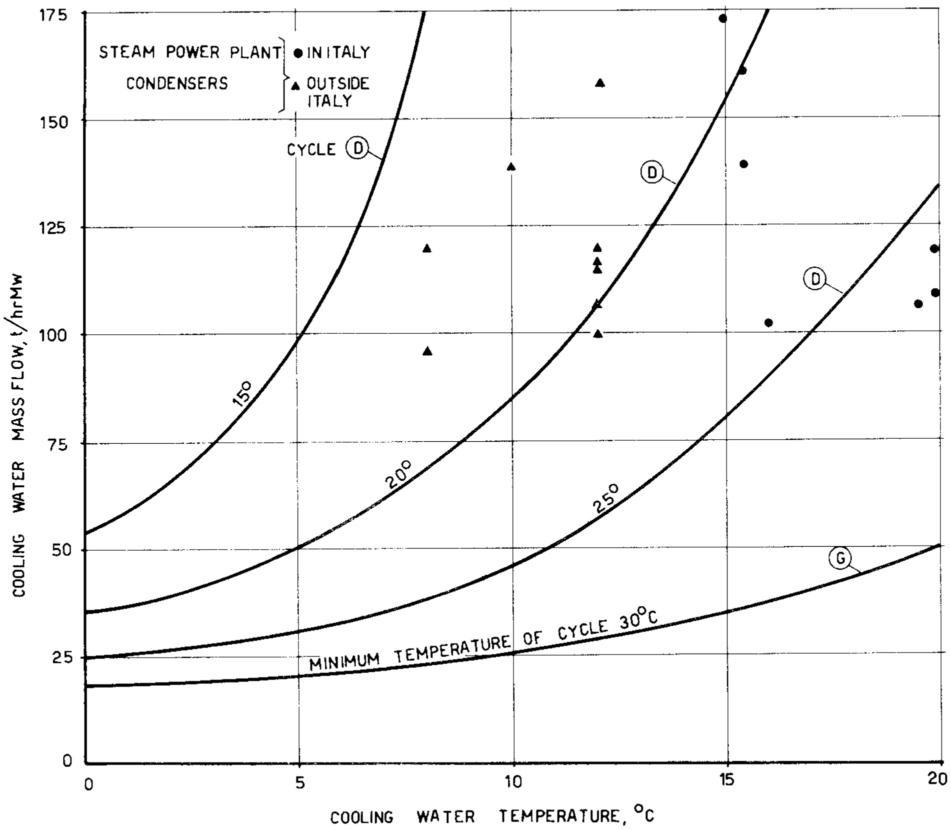


Figure 22

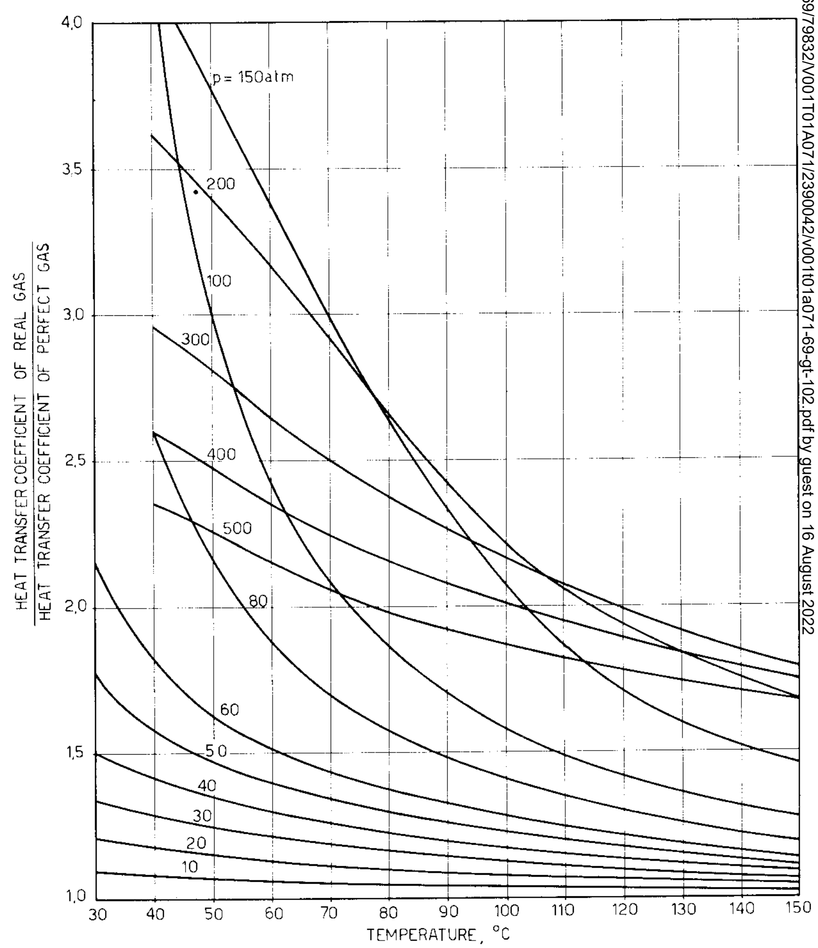


Figure 23