THE CRYO-THERMAL ENGINE UNDERWATER POWER SYSTEM: PERFORMANCES, DYNAMICS AND CONTROL

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1 ABSTRACT

A study on the applications of closed cycle diesel to power underwater vehicles in deep water led to the development of a new concept for exhaust gas treatment and management.
The totally closed system is based on liquefaction of combustion carbon dioxide after low pressure compression. Cooling is provided by the evaporation and superheating of liquid oxygen sent to the engine for combustion.

The design activity proved it to be efficient, simple and of minimum mass, particularly for very deep waters, and ultimately suitable for combination with a variety of thermal engines, including stirling.
The paper covers system's configuration, performances, volume and mass characteristics for the specific case of a long range autonomous ROV for deep ocean surveys.
Mention is also paid to preliminary analysis of its dynamic performance and control system.

2 INTRODUCTION

Autonomous underwater vehicles are proposed and are under development for a number of tasks, among which:

- surveys of large bottom areas for objects searching,
- detailed bathymetric surveys prior to installations of sealines or cables,
- subsea ice-keel profiling for arctic or antarctic surveys,
- mine neutralization
- bottom and sea surveillance and environmental monitoring,
- exploration of the ocean floor for scientific research and resources assessment.

All these tasks require long autonomies, say in excess of 500 km or 80 - 100 hours at a low average speed (1.5 - 3 m/s) thus weight-buoyancy balancing of the vehicle. The useful payload mass ranges in general 800 to 1000 kg, neutrally buoyant. The desirable water depth capability is 3000 m for Mediterranean operations and 6000 m for open ocean exploration.
Supplying power to these systems, in the order of tenths kW, is still object of development.
Electric rechargeable batteries, even of advanced performances, do not allow to achieve acceptable vehicle size.

Radioisotope Rankine cycle or thermoelectric generators are not suited for civil applications, due to the cost and the unacceptability of a potential uncontrolled loss of the vehicle and the fuel; for defence applications they are most suited for autonomies far higher than the figure given.
The most promising systems are therefore fuel cells and closed cycle thermal engines. Among the latter Stirling and closed cycle Diesel engines have a great potential where cost, safety and reliability play a major role in the vehicle design as well as energy density.
Recycle or closed cycle anaerobic thermal engines have been proposed since the end of last war in a number of solutions. In all cases the mixture is formed in a chamber up-stream the engine inlet where pure oxygen coming from a storage system is mixed in proper proportion with the engine exhaust gases, after cooling (Figg. 1 a,b)
The exhaust gases management systems were mainly conceived for the envisaged applications at water depths limited to few hundred meters.
They have each for a different aspect drawbacks, either in terms of depth capability and independence or of size or of thermal or noise tracing; reliability is instead still a qualitative issue, associated to solutions' complexity, since no sufficient record of operation exists to quantify it properly.
The object of this paper is an exhaust gases and oxygen management and storage system suitable for any kind of hydrocarbon combustion engine although its performance and its advantages have been evaluated particularly with diesel engines.

3 UNDERWATER THERMAL ENGINES AND THE GASES PROBLEM

3.1 Exhaust gases disposal

In all systems cited there is a defined rate of excess gas produced by the fuel combustion, whose main components are CO_2 and H_2O in stoichiometric proportion with fuel's formula. For engines burning diesel fuel, as is the most common case, CO_2 and H_2O produced correspond in mass respectively to about 90 % and 40 % of the oxygen burnt; residual carbon monoxide is negligible and does not affect the problem herewith dealt with, as well as NO_x products which would be produced only if nitrogen[1] is in the inert gas mixed with oxygen. Instead the presence of unburnt carbon parcticles and sulfur acids produced if the fuel is not of proper quality suggests using a water spray cooler rather than a surface cooler downstream the engine, to clean-up the recycled gas.
The various systems proposed differ principally in the way the excess gas problem is managed in strict dependence with the engine design and performance with the artificial mixture supplied. Two main classes of solutions can be identified.

Disposal outside the vehicle at hyperbaric pressure.

The excess water is simply condensed and stored onboard the vehicle in all solutions developed.
Compression outboard the system of the excess gas was proposed for the first experiences with closed cycle diesel (Puttick, 1971). This solution is limited to shallow water depths (e.g. 100 m) for power and overall efficiency reasons and entails noise and loss of a significant part of the oxygen stored and remained within the exhausts.
A more efficient solution, based on CO_2 absorption by seawater, has been developed by Santi (1983) and Fowler & Boyes (1987); these systems require to introduce sea water in the order of about 100 kg per kg of CO_2 to be exported if the absorber works at 4 - 5 bar absolute; subsequently this enriched water is to be pumped outside up to hyperbaric pressure. The designs proposed succeed in reducing to a minimum the power consumed for this process by exploiting the inlet water head to help the pumping action by means of volumetric systems and switch valves. However in very deep water the compressibility of this liquid becomes increasingly important and leads to increasing power wasted for the process. All these problems can be counterbalanced but at obvious expense of the simplicity and in the end of the cost effectiveness and reliability of the system.

[1] A closed cycle diesel or stirling engine for an ROV can most conveniently be designed to run only under closed cycle conditions thus avoiding any contact with air.

Treatment and storage inside the vehicle

Thompson & Fowler (1978 to 1981) and Haas (1988) have proposed systems based on CO_2 reaction with a solution of KOH, whereas Asada & Nagai (1969 to 1980) proposed absorption by a solution of Monoethanolamine (MEA). In both cases the vehicle must store on board a considerable additional load which reduces the specific energy and power of the unit to unattractive values.
For the two absorbers envisaged even assuming saturation concentration with the water solvent the additional mass to be transported, not accounting for the storage and circulation equipment, would amount to 2.3 to 2.5 kg per kg of oxygen burnt.
A lighter solution, whose principle is close to that described herewith, was proposed by Brunner (1974). It consists in the compression of the CO_2, which is the sole inert gas, at high pressure (between 60 and 80 bar), followed by sea water cooling. If the water temperature is low enough (< 15°C) cooling allows liquefying the CO_2 and recovering the uncondensated O_2, otherwise the excess gas is stored as high pressure mixture.

3.2 Storage and supply of the oxygen

At very high water depth and long autonomies the preferred solution is to store oxygen in liquid form to minimize the overall displaced volume and mass of the storage vessels, thus the energy and oxygen requirement on their own.
Therefore, in parallel with the problem of managing the exhaust gases, thermal engines require also to evaporate the oxygen while it is consumed and to heat it from about -180°C up to a temperature suitable for mixing with recirculated gas and manifolding (e.g. 0 to 10 °C).

3.3 Conclusion for long autonomy, deep water ROVs

The above discussion can be summarized as follows:

- totally closed systems are preferrable

- the engine can be a closed cycle diesel, preferably with CO_2 as the recirculated inert gas (Brunner's system) or stirling.

- liquid oxygen storage is preferrable.

4 THE CRYO-THERMAL ENGINE

4.1 Basic physical principle

The above background leads to the exhausts and oxygen management and storage system studied by the authors which optimizes the energy exchange to allow low temperature and pressure liquefaction of the CO_2 produced by means of the cooling power of the O_2 evaporating and heating-up. Its behaviour is practically independent from sea water conditions.
Fig. 2 shows the basic enthalpy data and transformations which allow the basic comprehension of the process by the stationary conditions equation:

(1) $$m_{ox} \Delta h_{ox} = m_{cd} \Delta h_{cd}$$

4.2 Description of the system and its process

Fig. 3 shows the process diagram of the CRYO-THERMAL ENGINE using a recirculated diesel engine, on the left side of the figure, working on a binary CO_2/O_2 mixture, discharging at low pressure (1 to 2 bar).
The right side of the system is instead a gas management system, composed of the following main components:

- two insulated storage vessels, the oxygen one being installed above the other for the carbon dioxide; inside this is installed the 2nd stage CO_2 condenser, cooled by oxygen;

- one liquifying stream piping with condensated water separator, compressor, post compression water-cooler, filters and dehydrators, counterflow cryogenic cooler and 1st stage condenser through which excess exhausts are flown;

- one oxygen delivery outlet piping which collects both the pure gas evaporated and heated up and residual incondensated gas coming from the CO_2 vessel;

The mass and volume of the CO_2 produced are of the same order of those of the consumed oxygen, allowing the use of similar or identical storage vessels for the two fluids, although separated.

The system performs the following functions:

- delivering pure oxygen at low pressure to the distribution manifold;
- maintaining constant pressure in the oxygen tank;
- liquefying CO_2 by oxygen cooling;
- maintaining constant pressure in the CO_2 tank by optimizing the venting flow rate according to the working conditions, to achieve minimum compression power consumption.
- recovering all residual oxygen.

The engine exhausts are cooled from about 450°C down to ca 20 °C by secondary fresh water, sprayed in counterflow to allow separation of solid or liquid parcticles trapped in the gas stream. Combustion water is condensed and separated; the gas, which contains a residual oxygen, is sent to a mixer where fresh oxygen is added by a regulator. A proper amount of the recirculated and dehumidified gas is deviated by an electrically powered compressor, activated by a pressure control loop in the water separator. Down-stream it, a dehydrating filter and a postcompression water cooler allow the compressed gas to enter the cryogenic section free of water and at the minimum temperature achievable with sea water.
The CO_2/O_2 mixture enters a surface heat exchanger in counterflow with oxygen vapor, which is superheated from ca -170 up to 0°C. On the other side CO_2 starts condensating and completes the process within the low temperature condenser and storage tank, going in contact with a second heat exchanger where liquid oxygen is evaporated. The flow in this evaporation loop is returned to the oxygen tank by natural circulation, thus avoiding cryogenic pumps in the system. The flow in this loop is controlled by a regulator keeping a constant gas gap pressure in the oxygen tank.
The residual gas incondensated in the CO_2 tank, which is an oxygen rich binary mixture, is vented to a manifold which collects both this and the pure oxygen flowing from the superheater. Venting is controlled by a regulation loop fed-back by the pressure and the temperature in the CO_2 tank.
Should the closed cycle diesel work on a ternary mixture, i.e. with Argon or Nitrogen and CO_2 as inerts, the system must be complemented with a low pressure gas separating unit (e.g. semipermeable membrane) to minimize presence of incondensable gases in the CO_2 tank, thus of total pressure and compression power.

4.3 Performances under stationary conditions

Appendixes report on the thermodynamic relationships which describe and quantify the balances of this process. Those relationships have been used as the basis for the preliminary design of a power system for an autonomous ROV powered by a CO_2/O_2 closed cycle diesel. The basic requirements of the power source are summarized in Tab.1.
The liquefaction system design data are instead reported in Tab. 2. Only two parameters are theoretically related to the energy/power rating of the system:

$$\frac{D_{ox}}{m_{ox}} \quad \text{and} \quad \frac{D_{cd}}{m_{ox}}$$

(tanks heat loss coeff./Oxygen flow rate)

Their value is however very small and practically ininfluent if the tanks are Dewar-like with superevacuated gap; the other parameters are pure physical characteristics. Under these conditions the performances shown apply with very good approximation to systems having any power rating and any energy autonomy.
Figs. 5 to 8 show the dependance of main system's parameters under stationary conditions from compressor's relative mass flow rate (f); the latter is the non-dimensional ratio between the actual flow rate and its theoretical minimum value which would deliver only the excess CO_2 to be condensed, pure O_2 resulting at the CO_2 vessel vent outlet.
It is evident that low temperatures can be reached, allowing low condensation pressure, even at fairly high values of f.
Condensation start temperature and total pressure show a minimum (35°C and 14 bar respectively) for

$$f \approx 1.3$$

Fig. 8 shows the compression power, as percentage of the shaft power available, in function of f for engine's specific fuel consumption of 240 and 300 g/kWh, corresponding to specific oxygen consumption of respectively 836 and 1045 g/kWh. It also shows a minimum for a lower value of f (1.1 ÷ 1.2), of ca 7 and 9 % in the two cases. These values account for a compressor efficiency of 65 %, suitable for low capacity units.

4.4 Dynamic performances and control

The Cryo-thermal closed cycle diesel of Fig. 3 has a total of five variables to be controlled; a preliminary system analysis has been performed considering independent PID control loops which resulted in the definition of the following types of regulators:

Parameter	Regulator type
- oxygen concentration in the mixer	PID
- compressor's flow rate	P
- CO_2 tank pressure	P
- Oxygen tank pressure	PI
- engine speed	PI

In perspective these control functions shall be integrated in a multivariable digital controller which will account for the system's non linearities. The system is indeed identified by 27 equations of as many variables. 275 linearization constants are defined whose value depend on the engine working conditions.
To evaluate the system's degree of non linearity the above constants have been calculated for a base case (average power condition) and for two extreme varied cases

Case 1: variation to max torque @ constant speed

Case 2: variation to max speed @ constant torque

Their resulting variations relative to the engine parameters (torque or speed) variations from the base case are shown in Tab. 3. It can be seen that some, but few, constants have a relative variation of the same order of that of the system working parameters. Therefore a more comprehensive control evaluation of the whole system is to be performed. The overall system analysis was, so far, limited to verifying the observability and controllability of the system, which resulted positive, and to evaluating the transition time constants dominating the liquefaction system dynamics.
This system can be regulated, as anticipated, by opening the relief valve V_i when the accumulation of incondensables tends to increase the total pressure, while it is closed when the pressure has reached or is close to the minimum. Monitoring of the temperature allows to solve any ambiguity due to the two values function.
However it is worthwhile mentioning that non stationary conditions studied with the full equation (8) are dominated by the time constant of the CO_2 and O_2 vessels which can be expressed as:

$$t_{st\ tank} = \frac{\text{Total tank enthalpy}}{\text{Net enthalpy flow rate}}$$

Their values, for the design considered, are in excess of two hours if one vessel is 5 % full and of the order of 20 hours if both are 50 % full. This allows to conclude that the system's working conditions are extremely stable even in case of temporary unbalance of the mass flows of CO_2 and O_2.
Furthermore the above allow to foresee, as a safe emergency procedure that the vehicle can and must surface to a depth where oxygen relief is possible, in case of failure of any part of the system which may cause its stop and result in the vessels shut-off. Before the thermal losses of the vessels bring the pressure to a design limit level, there is a far sufficient time to render possible and safe this procedure.
Alternatively both the CO_2 and Lox vessels should have their internal vessel resistant to the maximum navigation depth, to allow direct and automatic relief outboard, with obvious weight penalty.

5 COMPARISON WITH OTHER SYSTEMS PROPOSED FOR DEEP WATER VEHICLES

5.1 Comparison with other thermal engines

It is essential now to compare the performances of the system considering the net power loss from that available at the engine shaft. The reference condition is at -45 °C of final temperature condensation (ca 14 bar of total gas mixture pressure).
Fig. 9 shows the function of compression power, as percentage of the engine shaft power, at increasing engine's exhausts manifold pressure, thus at decreasing compressor's pressure ratios. Two curves are reported corresponding to the same values of s.f.c. and s.ox.c. of Fig. 8.
This parameter, for a given condensation temperature and total pressure, obviously decreases considerably if the engine can work under supercharged conditions.
Considering the same manifold conditions of the closed cycle diesel system described by Fowler & Boyes (1987) which works at 2 bar absolute pressure, the liquefaction system here described would consume between 5 and 7 % of the engine shaft power; this figure is similar to the CO_2 absorber performance reported by those authors, say 6% of max engine power which is however reported to increase remarkably at reduced load factor, unlike the system here described. The comparison shows an efficiency advantage of the liquefaction system if very deep waters are considered; indeed, as long as depth increases the effects of sea water compressibility becomes increasingly important and penalizes all systems based on CO_2 absorption by sea water at low pressure inside the vehicle. At 6000 m water depth sea water expands of about 3 % while it is introduced in the low pressure absorber, requiring an additional 4 to 5 % of the available power to be pumped out after the absorption of CO_2. The total figure of the parasitic losses due to closed cycle operation would then increase for those systems to 11 %, or more at low load factor.

The system here described would instead maintain its performance characteristics as it is totally closed.
A further advantage of liquefaction systems over open absorber systems is the higher level of reliability that can be achieved, since neither very high pressure valve for water management nor water ballast system for exhausts mass disposal compensation are necessary.
As to closed cycle diesel systems based on absorption by KOH or MEA, the comparison is mainly in terms of specific energy and power achievable (Wh/kg and W/kg respectively) and is definitely in favour of the liquefaction process due to the lower mass and volume of reactants and products. Closed cycle diesel systems, like Brunner's (1974), based on high temperature and pressure liquefaction of the CO_2 compare with the cryogenic by higher power losses (15 to 20 % of engine's output) and by heavier heat exchangers for the lower temperature difference available.
It is also evident that the liquefaction system can be combined advantageously with a Stirling engine; in this case the power loss would be nil as the condensation total pressure is below the combustion chamber pressure (Nilsson, 1987, Sauzade, 1988) of 20 bar (see Fig. 9).

5.2 Comparison with other underwater power sources

The Cryo-thermal diesel system designed under the specifications of Tab. 1 has a total mass of ca 5500 kg, including reactants. Out of this mass ca 1/3 is due to power dependant equipment (engine, compressor, piping, heat exchangers, valves, etc.and relevant pressure vessels) and 2/3 for components dependant on energy autonomy (reactants and their and products' pressure vessels). Therefore the characterizing specific parameters at 6000 m water depth are 160 kg/kW and 6.2 kg/kWh, to be used in the pre-dimensioning equation

$$M = 160\ P_{kW} + 6.2\ E_{kWh} \quad \text{(kg)}$$

The resulting unit is slightly positively buoyant.
Figs. 10 and 11 give a comparison, respectively in terms of mass-buoyancy and cycle-cost, under the same specifications, of the Cryo-thermal diesel engine with batteries, fuel cells based on liquid Hydrogen and Oxygen and c.c. diesel based on sea water liquefaction.
It is evident the far long advantage over battery systems, while the fuel cells considered have a mass advantage which shall be however regarded in comparison with the known logistic and reliability problems they pose.

6 CONCLUSIONS

The above discussion demonstrates that a very deep water power source based on either diesel or stirling engine is feasible, efficient and competitive in terms of cost and weight with the other systems having similar safety caracteristics for the personnel, in all operative conditions.
The described process, based on the liquefaction of the excess water and CO_2 produced by HC fuel combustion is the most attractive, featuring the following advantages:

efficiency: process power losses are low; with supercharged engines featuring a s.f.c. of 240 g/kWh they are 5 % of engine shaft power at any water depth and load conditions; this due to the CO_2 liquefaction at low temperature (-45 °C) and total gas pressure (ca 14 bar);

simplicity and reliability: diesel engines have a long track of reliability and work even better with cleaned recirculated gas; interfaces with the environment are minimum and no moving components act against ambient water head; no cryogenic pump is necessary in the system;

weight & volume minimization: a system designed for 6000 m w.d., 12 kW max power and 600 kWh energy autonomy has a mass of ca 5500 kg, split as 160 kg/kW and 6.2 kg/kWh, including pressure resistant vessels and reactants; this system has a residual reserve buoyancy;

full depth independence: apart from the weight all the above performances are maintained at any depth thanks to the totally closed process;

noise and heat signature minimization: unlike open systems, a closed cycle heat engine has no gas venting ports which generate noise and high temperature traces; for applications where indiscretion is of primary importance in some part of the mission, integration of c.c. diesel with advanced batteries allows maintaining the advantages above mentioned.

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List of symbols		Units
c_p	constant pressure specific heat @ T	kJ/kg°C
$c_p\|_{T_1}^{T_2}$	constant pressure average specific heat between T_1 & T_2	kJ/kg°C
c_v	constant volume specific heat @ T	kJ/kg°C
$c_v\|_{T_1}^{T_2}$	constant volume average specific heat between T_1 & T_2	kJ/kg°C
g	mass concentration in a mixture	kg/kg
h	specific enthalpy	kJ/kg
m	mass flow rate	kg/s
p	pressure	N/m^2;bar
u	specific internal energy	kJ/kg
t	time	s
x	mole concentration in a	
B_f	CO_2 produced per O_2 burnt (fuel constant)	kg/kg
D	= kS, thermal loss coefficient x surface	kW/°C
M	total mass in one vessel	kg
N_{st}	number of compressor's stages	
Q	thermal loss of one vessel	kW
T	temperature Celsius	°C
U	total internal energy in one vessel	kJ
f	mass flow ratio	
ß	stage compression ratio	
η_c	compressor efficiency	
μ	molecular weight	

Indexes

cd	carbon dioxide
cds	carbon dioxide storage vessel
dcd	difference (in CO_2 evaporation enthalpy)
e	external (sea water)
i	total (incl. CO_2) incondensated gases in m_i
i	incondensable gases (O_2) in g_i,x_i
ox	oxygen
oxs	liquid oxygen storage vessel
DE	dehydrated exhausts
1÷4	flow sections (see Fig. 4)

APPENDIX

A.1 Thermodynamic and mass balances definitions

Fig. 4 shows the boundaries of the semiclosed system which includes the O_2 and CO_2 vessels and exchanges mass with the environment by three in/outlet streams:

m_{ox} pure oxygen mass flow rate from the LOX vessel (negative[1])

m_{DE} Dry Exhaust mass flow rate from the compressor (positive)

m_i incondensated gases mass flow rate from the CO_2 condenser and storage vessel (negative).

The first flow rate is determined by the engine load demand, the second is generated by the compressor which is run at varying speed or on/off to maintain constant pressure in the engine closed circuit, the third is generated by the control valve V_i.

A.2 Mass balances equations

A first relationship is the continuity equation for the incondensables in the CO_2 vessel:

$$g_{i4}\, m_i = - g_{i1}\, m_{DE} \tag{2}$$

where $g_{i1,4}$ are the mass concentrations of these gases, mainly oxygen, in sections 1 and 4 respectively.
Another applicable relationship is the CO_2 balance or the stoichiometric relationship with the oxygen actually burnt by the engine:

$$m_{DE} + m_i = - B_f\, m_{ox} \tag{3}$$

where B_f is a constant dependent from the H/C ratio of the fuel and represents the mass of CO_2 produced per unit mass of O_2 burnt; for diesel fuel

$$B_f = 0.892$$

Both eqs. (2) and (3) hold true over a sufficiently long time interval, as the result of the system's control to maintain constant pressure in the closed cycle engine system and in the CO_2 vessel.
It is useful to define the mass flow ratio

$$f = \frac{m_{DE}}{m_{DEmin}} > 1 \tag{4}$$

[1] In the formulas the following convention has been assumed: flows entering the system are positive

where m_{DEmin} is the theoretical minimum value which would correspond to $g_{i4} = 1$ and to the min value of m_i according to eq. (3) for given values of the <u>engine working parameters</u> m_{ox} and g_{i1}

$$(5) \qquad m_{DEmin} = - \frac{B_f\, m_{ox}}{1 - g_{i1}}$$

The above relationship imply that the concentration g_{i4} is defined univocally as function of f and g_{i1}:

$$(6) \qquad g_{i4} = \frac{f\, g_{i1}}{f - 1 + g_{i1}}$$

A.2 <u>Energy balance equation</u>

With the symbols used in Fig. 4 the following balance equation apply:

$$(7) \qquad m_{DE}\, h_{DE1} + m_{ox}\, h_{ox3} + m_i\, h_{i4} + Q_{oxs} + Q_{cds} = \frac{d}{dt}\, (U_{oxs} + U_{cds})$$

With the definitions, expansion and simplification reported in A.4 the following equation can be derived:

$$(8) \qquad - M_{oxs}\, c_{vLox} \frac{dT_{Lox}}{dt} + m_{ox}\, (h_{ox3} - u_{oxs}) + D_{ox}\, (T_e - T_{Lox}) = M_{cds}\, c_{vLcd} \frac{dT_{Lcd}}{dt} - m_{DE}\, (h_{DE1} - u_{cds}) - m_i\, (h_{i4} - u_{cds}) - D_{cd}\, (T_e - T_{Lcd})$$

Eq. (8) allows examining both time varying conditions and stationary conditions.
In the latter, which is worthwhile to examine here in detail, the two time derivatives are nil as is the case when a control system maintains constant the pressure in the vessels thus both T_{Lox} and T_{Lcd}.
In the system considered this is done by the valves V_{Lox} which opens to send liquid oxygen to the evaporator when the pressure in the LOX vessel decreases as consequence of m_{ox}. The action of the valve V_i is similar, being opened when the pressure in the CO_2 vessel increases above the set level, both as consequence of incondensables accumulation and of m_{DE}.
Under these stationary conditions eq. (8) becomes:

$$(9) \qquad m_{ox}\, (h_{ox3} - u_{oxs}) + D_{ox}\, (T_e - T_{Lox}) = - m_{DE}\, (h_{DE1} - u_{cds}) - m_i\, (h_{i4} - u_{cds}) - D_{cd}\, (T_e - T_{Lcd})$$

In eq. (9) the stream section 4 is relative to a mixture of gases, namely oxygen or in general incondensables with mass ratio g_{i4} and CO_2 with mass ratio $(1-g_{i4})$.
The same applies to section 1 with the obvious symbol notation.
Considering the compounded expressions of the enthalpy in 1 and 4, and the continuity equations (2) and (3), eq. (9) can be rewritten in the final form as:

$$(10) \qquad h_{ox3} - u_{oxs} + \frac{D_{ox}}{m_{ox}}\, (T_e - T_{Lox}) = f\, B_f\, [(h_{cd1} - h_{vcd}) + \frac{g_{i1}}{1 - g_{i1}}\, c_{piox}|_{T_{Lcd}}^{T_1}\, (T_1 - T_{Lcd})] + B_f\, h_{dcd} - \frac{D_{cd}}{m_{ox}}\, (T_e - T_{Lcd})$$

where the mass flow parameter f allows considering different situations of absolute flows but normalized to the unit flow of oxygen exiting the system. In eq. (10), among the various symbols, "h_{dcd}" is the evaporation enthalpy of the CO_2 at the vessel temperature T_{Lcd}.
Eq. (10) allows solving the design of the liquefaction system and its optimization.
In fact liquid oxygen storage parameters are usually fixed (pressure and temperature) or can be fixed as the set point of the Lox control system. Also the desired value for its output temperature can be fixed as a design parameter, whereas the inlet enthalpy of the dehydrated gases is imposed by both the engine working condition, which determines g_{i1}, and by the minimum achievable temperature through the water cooled heat exchanger downstream the compressor.
In the end all the remaining variables of the design problem in eq. (10), namely the enthalpies at various states of the CO_2 are direct function of the temperature T_{Lcd} except the mass flow parameter f which can be chosen to optimize the power consumption of the compressor as will be shown in the following.
It is useful to solve eq. (10) in terms of f, which must satisfy also the condition $f>1$; this leads to equation (11) herebelow.

The last relationships which must be considered for the design are those which define the total pressure in the CO_2 vessel, by the law of partial pressures (eq. 12) and the compression work corresponding to m_{DE} and P_{Tcds} (eq. 13).

In conclusion eqs. (4), (5), (6), (11), (12) define the variables m_{DE}, m_i, g_{i4}, T_{Lcd}, P_{Tcds} as function of the engine working parameters m_{ox} and g_{i1}, while eq. (13) allows calculating the compression work.

$$(11) \qquad f = \frac{h_{ox3} - u_{oxs} - B_f\, h_{dcd} + \frac{D_{ox}}{m_{ox}} (T_e - T_{Lox}) + \frac{D_{cd}}{m_{ox}} (T_e - T_{Lcd})}{B_f \left[h_{cd1} - h_{vcd} + \frac{g_{i1}}{1 - g_{i1}} c_{piox} |_{T_{Lcd}}^{T_1} (T_1 - T_{Lcd}) \right]}$$

$$f \geq 1$$

$$(12) \qquad P_{Tcds} = \frac{P_{vcd}\ (T_{Lcd})}{1 - x_{i4}} \qquad \text{where} \qquad x_{i4} = \frac{\frac{g_{i4}}{\mu_i}}{\frac{g_{i4}}{\mu_i} + \frac{1 - g_{i4}}{\mu_{cd}}}$$

(x_{i4} = incondensables mole fraction)

$$(13) \qquad \frac{W_c}{- m_{ox}} = \frac{1}{\eta_c} N_{st} \frac{k_m}{k_m - 1} P_c v_c \frac{B_f}{1 - g_{i1}} f \left[\beta^{\frac{k_m - 1}{k_m}} - 1 \right]$$

$$\text{where } \beta = \left[\frac{P_{Tcds}}{P_c} \right]^{\frac{1}{N_{st}}}$$

and N_{st} is the number of compression stages.

A.4 Notes

1) Eqs. applicable from eq. (7) to eq. (8):

$$\frac{d}{dt} (U_{oxs} + U_{cds}) = m_{ox} u_{oxs} + M_{oxs} \frac{du_{oxs}}{dT} \frac{dT_{Lox}}{dt} +$$

$$+ (m_{DE} + m_i) u_{cds} + M_{cds} \frac{du_{cds}}{dT} \frac{dT_{Lcd}}{dt} ;$$

$$\frac{du_{oxs}}{dT} = c_{vLox} ; \quad \frac{du_{cds}}{dT} = c_{vLcd} ;$$

$$Q_{oxs} = D_{ox}(T_e - T_{Lox}) ; \quad Q_{cds} = D_{cd}(T_e - T_{Lcd})$$

2) Eqs. applicable from eq. (9) to eq. (10):

$$h_{i4} - u_{cds} = g_{iox4} h_{iox4} + (1 - g_{iox4}) h_{vcd4} - u_{cds}$$

where $u_{cds} \approx u_{Lcd} \approx h_{Lcd}$ (very good approx. for vessels filling ratios above ca 10%), thus

$$h_{i4} - u_{cds} = g_{iox4}\ (h_{iox4} - h_{vcd4}) + h_{dcd} ;$$

$$h_{DE1} = g_{i1}\ h_{iox1} + (1 - g_{i1})\ h_{cd1} ;$$

$$h_{iox1} - h_{iox4} = c_{piox} |_{T_{Lcd}}^{T_1} (T_1 - T_{Lcd})$$

In the above the subscript "ox" stands more generally for an incondensable gas mixture of compounded characteristics (for the example shown relative to a CO_2/O_2 closed cycle the incondensables reduce to the sole oxygen).

USEFUL PEAK POWER	12	kW
USEFUL AVERAGE POWER	7,5	kW
MINIMUM CONTINUOUS POWER	2,5	kW
USEFUL ENERGY AUTONOMY	600	kWh
MAX OPERATIVE WATER DEPTH	6000	m

Tab. 1 - Energy and power requirements for an autonomous ROV

Exhausts incondensables mass concentration	g_{iox1}	8,84	%
External water temperature	T_e	10	°C
Dehydrated exhausts temperature	T_{DE}	20	°C
Oxygen delivery temperature	T_{ox3}	0	°C
Liquid oxygen temperature	T_{LOX}	-173	°C
Vessels unit heat	$\frac{D_{ox}}{m_{ox}}$	-0.02	$\frac{kJ}{kg°C}$
losses coefficient	$\frac{D_{cd}}{m_{ox}}$	-0.02	$\frac{kJ}{kg°C}$

Tab. 2 - Exhausts liquefaction system design data

Total n. constants having variation in the range	**Case 1**	**Case 2**
0.7 < Rel. var. ≤ 0.75	2	-
0.5 < Rel. var. ≤ 0.7	2	1
0.35 < Rel. var. ≤ 0.5	1	-
0.3 < Rel. var. ≤ 0.35	-	13
0.15 < Rel. var. ≤ 0.3	38	34
0 < Rel. var. ≤ 0.15	232	227

where $$\text{Rel. var.} = \frac{\dfrac{C_{case} - C_{base}}{C_{base}}}{\dfrac{Par_{case} - Par_{base}}{Par_{base}}}$$

Engine Case	Power (kW)	Torque (Nm)	RPM
Base	7.0	24.2	2760
1	10.8	37.2	2760
2	9.1	24.2	3600

Tab. 3 - System's non linearity evaluation; constants relative variation from base case to cases 1 & 2

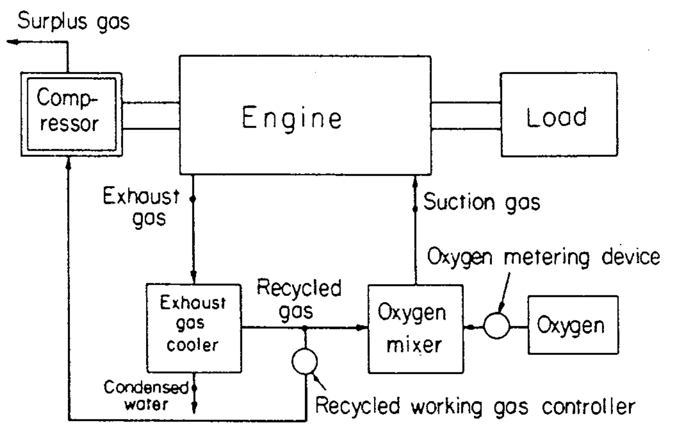

Fig. 1-a Recycle diesel engine
(Asada & Nagai, 1980)

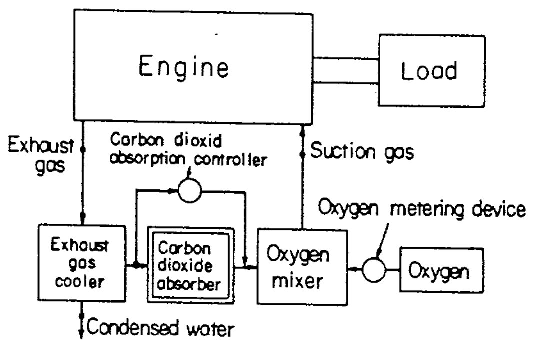

Fig. 1-b Closed cycle diesel engine (ibid.)

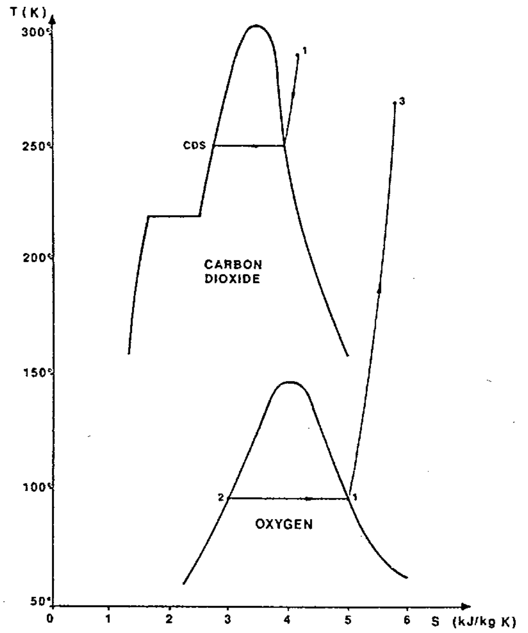

Fig. 2 Temperature - Entropy diagrams for CO_2 and O_2

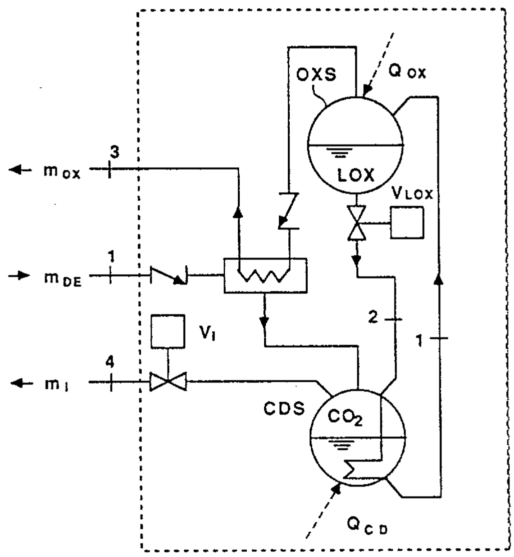

Fig. 4 Process box, mass and enthalpy balance

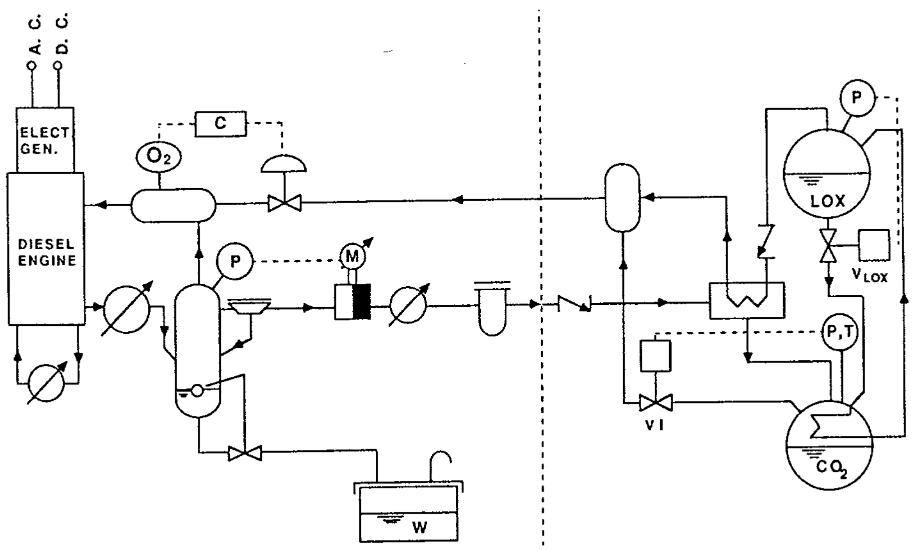

Fig. 3 Simplified diagram of closed cycle diesel engine with exhausts cryogenic liquefaction

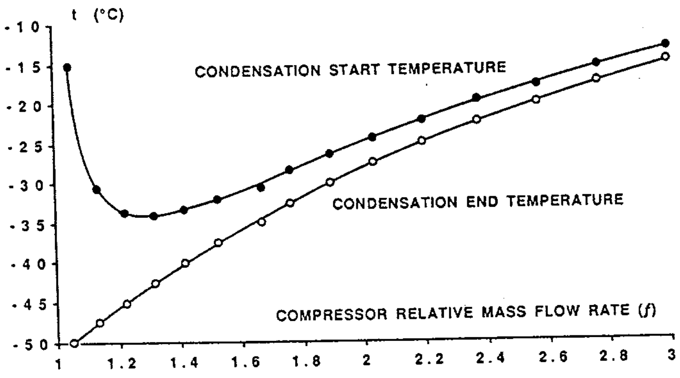

Figure 5

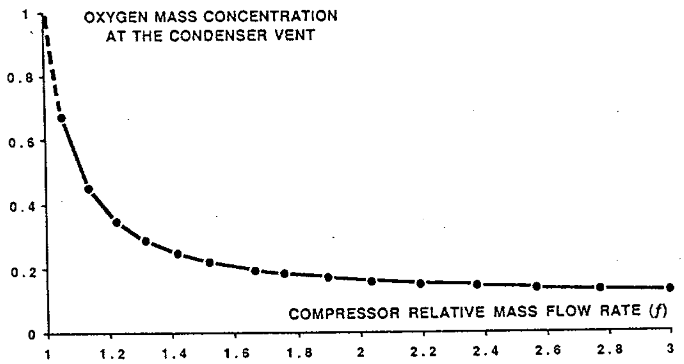

Figure 6

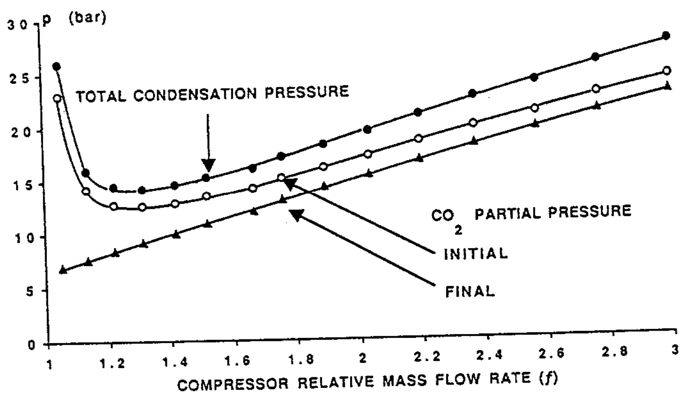

Figure 7

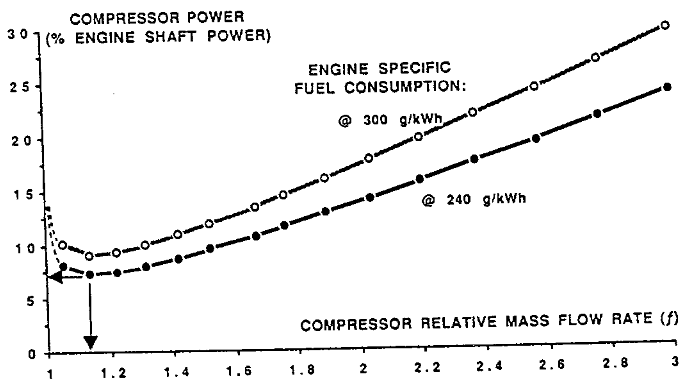

Figure 8

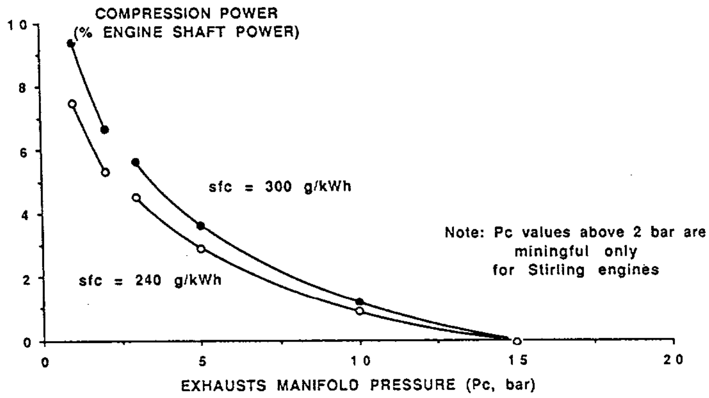

Figure 9

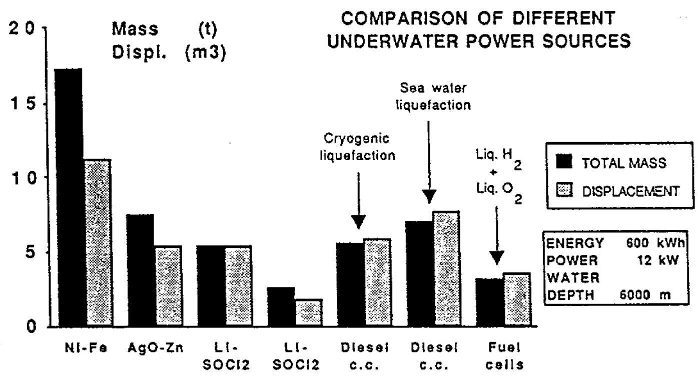

Figure 10

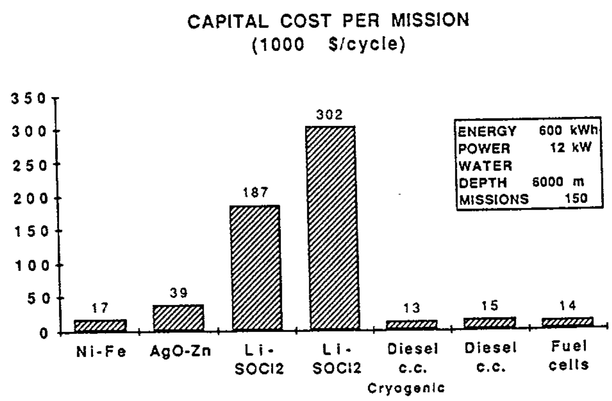

Figure 11