Analysis of a Wind Turbine Power Transmission System with Intrinsic Energy Storage Capability

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ABSTRACT
A wind turbine transmission system is described wherein mechanical power directly from the slow rotation of the shaft of a large wind turbine rotor is carried over to electrical power through a synchronous generator via the circulation of a high pressure gas running in a closed circuit. In the most straightforward mode of operation, power is injected into the gas circuit via special low-speed nearly-adiabatic compressors with very high isentropic efficiency and is extracted using an expander that is also nearly-adiabatic with relatively high isentropic efficiency. In other operating modes, it is possible to exploit the temperature changes arising naturally from adiabatic compression or expansion of gas so as to put energy into storage or recover energy from storage. This paper explores some of the design rationale behind such a power transmission system and uses exergy analysis to explain and evaluate its operation.

Keywords: Wind turbines, energy storage, transmission systems, thermal exergy storage.

1. INTRODUCTION
Wind turbines have evolved technically to the point where the power produced is competitive with that available from fossil-fuelled thermal generation in many contexts and the primary barrier to the wholesale penetration of wind energy into the electricity generation is ultimately connected with the capability to store energy. The ideal wind energy harvesting system would provide the capability to emulate a gas-fired power station in all respects excepting the emission of combustion products and the requirement to deliver fuel. The system studied here approaches that capability. In fact many possible system designs potentially have that capability and all necessarily incorporate some energy storage. It is a major challenge, however, to devise a design that can deliver the energy storage with good performance across the entire range of frequencies relevant to the electrical grid and at an affordable cost. This paper studies one option that could be realised in virtually any location where wind turbines might be sited.

Gas-fired generators provide energy storage over a wide range of frequencies. The natural inductance of the generator coils, transformers and transmission lines together provides enough energy storage to deal with disturbances to power demand at frequencies much higher than the synchronous transmission frequency. The physical inertia of the generator and the coupled turbine typically provide for frequency components ranging from ~0.1Hz up to low multiples of synchronous frequency. Some additional energy storage is present in the form of heat in the combustion chamber and steam in the boiler tubes (in the case of combined-cycle plant) and these also can contribute energy storage in the order of ~0.1Hz. At frequencies lower than ~0.1Hz, variations in power demand from a gas turbine generator are met by a control system acting to

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increase or decrease fuel burn rate. The ability of the plant to deal with these low-frequency components of power variation depends ultimately on the availability of gas storage. Evidently the power-generation machinery itself naturally provides much of the valuable energy storage capacity. The value of this service is realised increasingly today particularly as photovoltaic generation penetrates strongly into the electricity supply of many countries since the straightforward implementations of photovoltaic systems do not provide *system inertia*. Wind generation is presently treated in the same way – as *system non-synchronous* generation. The first and most important attribute of the system described here is that it transforms the wind powered plant into a power-plant having a synchronous generator with real inertia such that this system naturally provides energy storage to cater for all high-frequency (>~0.1Hz) variations in power demand. This answers some of the most immediate concerns affecting the extensive penetration of wind power into weak power systems [1].

The features provided by the power transmission system of interest here are:

1. It uses direct drive power take-offs from the wind turbine rotors – employing machines lighter and smaller than direct-drive synchronous or permanent-magnet generators having the same torque and power ratings.
2. It achieves energy storage using minimal additional power-conversion machinery other than that used anyway in the power transmission between the turbine rotor and the generator terminals.
3. It delivers an acceptable straight-through transmission efficiency from mechanical power at the turbine rotor to electrical power output at the generator terminals.
4. It enables effectively unlimited energy storage capacity to be added at very low marginal costs compared with other energy storage options.
5. It achieves high effective turnaround efficiency in the energy storage.
6. It does not require wholesale re-engineering of the wind turbine design.

These attributes can be held by a power transmission system in which the wind turbines drive compressors to circulate a gas in a closed circuit. Requirement 1 and 6 together demand that fluid pressures are reasonably high at every point in the gas circuit. Section 2 of this paper briefly explains the lower bound on pressure of the low pressure side of the closed gas circuit. This lower bound is based on the ability to fit a direct-drive compressor into the nacelle of a wind turbine and the affordability of that compressor. Requirements 2 and 4 are met by using a gas as the power transmission fluid and by implementing the energy storage in the form of thermal storage. Section 3 of the paper begins by identifying five distinct operating modes of the power transmission system and then outlines how the energy storage can work. This section quantifies how large the thermal stores must be for a given capacity of energy storage. Temperature considerations for the thermal energy storage play a strong role in determining the ratio of pressures between the high pressure and low pressure sides of the closed gas circuit.

Section 4 of the paper considers the viability of power transmission over distances in the order of 1km which could enable all of the wind turbines in a small group to share common thermal storage provisions and a common expander-generator set. This arrangement is likely to be preferable in many onshore applications. The efficiency requirements place restrictions on what pressure drops are tolerable in the tubes connecting each wind turbine to the centre and these pressure drops determine minimum pipe bores for a given set of pressures. It is seen that increasing the pressures in the closed gas circuit leads to a decrease in the material required for the pipework – with corresponding advantages in both cost and thermal inertia.

Great care is required in the design of the compressor and the expander if the high efficiencies needed to satisfy requirements 3 and 5 are to be achieved. The primary compressor to be fitted directly in the nacelles of wind turbines is an unusual machine. Section 5 discusses the design of this primary compressor with a view to indicating that the efficiency targets can indeed be met by at least one possible design. A short commentary on the expander is also presented in this section. Section 6 undertakes an analysis of what exergy losses can be expected realistically through imperfect or undesirable heat transfer. Some other practical issues associated with the system design are discussed in section 7. Finally, in section 8 there is a discussion of the potential commercial justification of power transmission systems such as the one studied – given that the cost per unit output power is likely to be greater than that of conventional wind turbine transmission systems.

2. POWER CONVERSION FROM WIND TURBINES BY COMPRESSING AND EXPANDING GAS

For any ideal gas being compressed or expanded in a polytropic process,

$$pV^n = \text{constant}. \tag{1}$$

If a machine inducts a volume V_1 of gas at pressure p_1, then compresses (or expands) it to a second pressure p_2 via a polytropic process and finally discharges the gas at that second pressure, the work done by the machine on the gas is

$$W = p_1V_1\left(\left(\frac{p_2}{p_1}\right)^{((n-1)/n)} - 1\right)\Big/\left((n-1)/n\right) \tag{2}$$

for all $n > 1$. In the specific case where $n = 1$, the work done on the gas becomes

$$W = p_1V_1 \log(p_2/p_1). \tag{3}$$

If we are designing a compressor to fit within a wind turbine nacelle and to be driven directly by the rotor, then V_1 will represent the volume of gas inducted by that compressor in each cycle, W will represent the work delivered from the wind turbine rotor in one cycle and p_1 will represent the pressure of the gas inducted. The work delivered per cycle is obviously 2π times the mean torque. The mean torque is proportional to the square of the rated wind speed, v_R, and the cube of blade-tip diameter, D_T, so that for some constant a, we have

$$W = a{v_R}^2{D_T}^3. \tag{4}$$

For a typical modern large (offshore) wind turbine with blade tip speed ratio of 10 and a rotor power coefficient of 0.475, $a \cong 73 \times 10^{-3}$ (Js^2/m^5). Note that the work per cycle is proportional to D_T^3 so that the required intake volume of gas per cycle is also proportional to D_T^3 if the two pressures, p_1 and p_2, are fixed. Table 1 shows the required intake volume per cycle for several combinations of intake pressure, p_1 and pressure-ratio (p_2/p_1) for a turbine with rated wind speed v_R = 11.05 m/s and blade tip diameter D_T = 126 m. The power of this turbine would be 5MW. In Table 1 it is assumed that the compression is completely adiabatic and that the gas is diatomic so that $n = \gamma = 1.4$.

Table 1 contains a number of impractical combinations of (p_1, p_2). One limitation on the operation relates to temperatures. During an adiabatic compression process following (1), the temperature rise is calculated using

$$(T_2/T_1) = (p_2/p_1)^{((\gamma-1)/\gamma)}. \tag{5}$$

With $\gamma = 1.4$, a pressure ratio of 25 will deliver a temperature ratio of 2.509 – meaning that if the intake gas arrives at 300K, it will leave the compressor at (a minimum of) 752.5K. For ease of

Table 1. Swept volume (m^3) per cycle for a 5MW direct-drive compressor with polytropic coefficient n = 1.4

	Pressure ratio (p_2/p_1)								
	3	**5**	**8**	**12**	**17**	**25**	**40**	**60**	**100**
p_1 = 1 bar	138.6	87.6	63.0	49.4	41.0	33.9	27.4	23.0	18.7
p_1 = 2 bar	69.3	43.8	31.5	24.7	20.5	16.9	13.7	11.5	9.37
p_1 = 5 bar	27.7	17.5	12.6	9.89	8.20	6.78	5.47	4.60	3.75
p_1 = 10 bar	13.9	8.76	6.30	4.94	4.10	3.39	2.74	2.30	1.87
p_1 = 20 bar	6.93	4.38	3.15	2.47	2.05	1.69	1.37	1.15	0.937
p_1 = 50 bar	2.77	1.75	1.26	0.989	0.820	0.678	0.547	0.460	0.375

thermal storage, an upper limit of <900K is reasonable. In the system of interest, the irreversibility (due to mechanical losses and thermal leakage) in the system will result in gas leaving the compressor hotter than equation (5) indicates. This supposes that the same pressure ratio (p_2, p_1) is achieved so that the decrease in efficiency results in reduced volume throughput. It also assumes that the compressor will be insulated to allow almost zero net heat to escape to ambient. Such insulation is helpful to the performance of the system described here.

Table 1 shows that if the power of a 5MW wind turbine was to be absorbed by adiabatic compression of dry ambient (~1bar) air in a direct-drive compressor, the intake swept volume of that compressor would have to approach 34 m^3 if the pressure ratio (p_2, p_1) remains below 25. The required intake swept volume is reduced only slightly if intercooling is introduced between compression stages such that the same temperature ratio is achieved with a much higher pressure ratio. For example, in an arrangement where one stage with a pressure ratio of 25 is followed by cooling the gas back to ambient temperature and then a further stage with a pressure ratio of 25, the second compression stage would absorb only (1/25) of the power that was absorbed by the first stage. In short, considering multi-stage compression does not solve the problem of huge intake swept volume. The problem of high intake swept volume arises because the pressure p_1 is small. To address this, we design the system for a higher value of p_1. This solution carries the price that there must be a closed circuit providing both an outward path from compressor to expander and a return path back from expander to compressor. It also means that the storage of varying quantities of pressurised air is no longer an option. Thermal energy storage becomes the main option for energy storage then and this provides a strong motivation for maintaining the pressure ratio (p_2, p_1) relatively high. There is no hard upper limit on the gas pressure in the closed circuit.

3. THERMAL ENERGY STORAGE INTEGRATED WITH THE TRANSMISSION

The change in temperature of a gas following adiabatic compression/expansion can be used to equip the power transmission with capability to store energy or to recover energy from storage. For an ideal gas, equation (2) shows that the power drawn by an adiabatic compressor is proportional to the intake volume flow rate. Similarly, the power returned by an adiabatic expander is proportional to its intake volume flow rate. Since the mass flow rate of gas around a closed circuit must be constant at all points in the circuit in steady state, the intake volume flow rates are proportional to temperatures. The system can be used to store energy by cooling the gas after compression (and storing the heat) and then by removing and storing coolth (coldness) from the gas after the expander. These temperature changes cause the compressor to draw much more work than the expander delivers. Conversely, the system can be used to recover energy from storage by adding further heat to the gas after the compression process and by adding coolth to the gas after the expansion. Then the expander delivers much more power than the compressor draws. This section provides some formal analysis for the five different operating modes of the system:

- Mode A: Direct power transmission from primary compressor to the expander
- Mode B: Transmission from primary compressor to expander with some energy going into storage
- Mode C: Transmission from primary compressor to expander with some energy returning from storage
- Mode D: Power insertion via secondary compressor towards expander with some energy going into storage
- Mode E: Power insertion via secondary compressor towards expander with some energy returning from storage.

Note that modes B and D can be combined in any proportion desired. Similarly modes C and E can be combined in any proportion desired. Note also that if the system is operated in modes D and E only, it is effectively a standalone energy storage provision that can take electrical energy off the grid and restore (a fraction of) this energy to the grid.

Figure 1 provides a schematic of the proposed transmission system showing several wind turbines at the top of the diagram – each connected to a primary compressor. These primary compressors draw gas from the low pressure and low temperature side (left) of the closed gas circuit and drive it over to the high pressure and high temperature side (right) after compressing it.

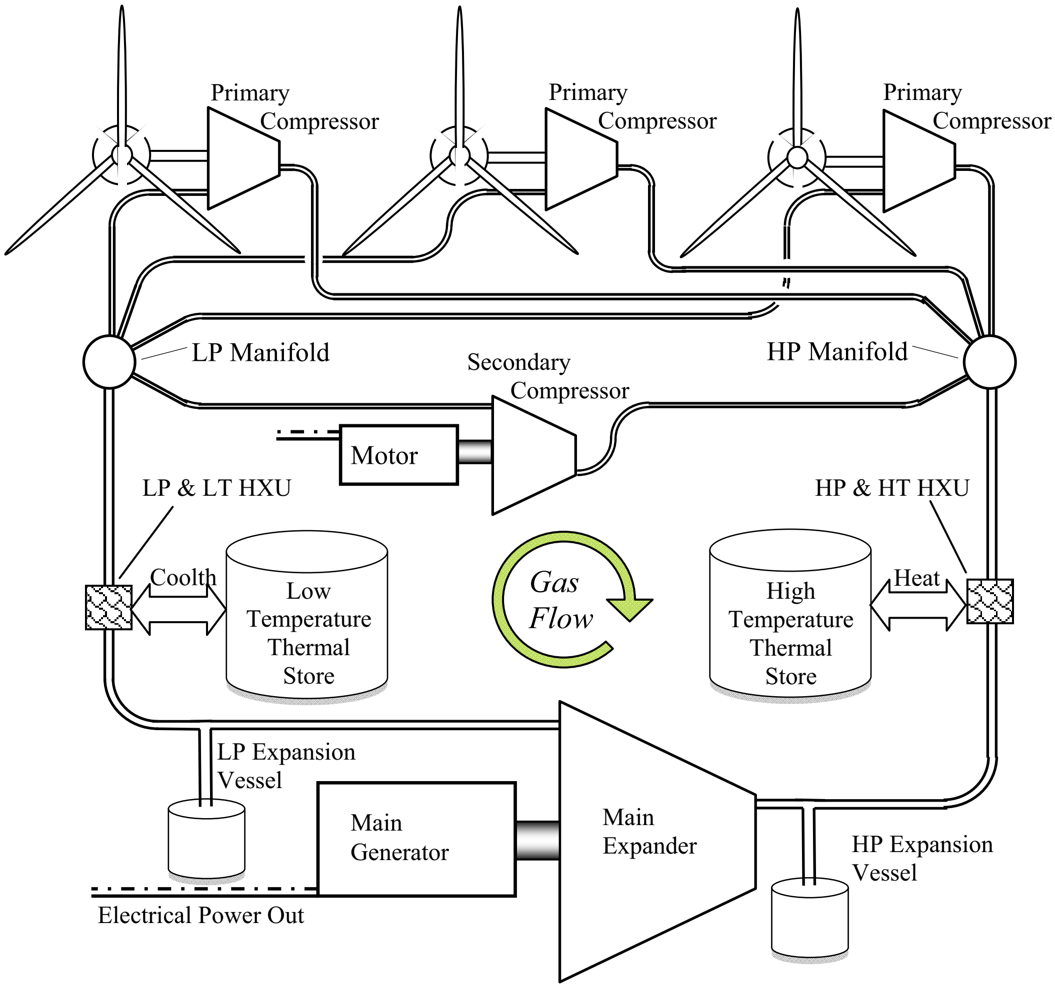

Figure 1. Schematic of the power transmission system of interest

The main expander-generator provision is shown at the bottom of the diagram. A secondary compressor provides a parallel path for gas to be driven from the low pressure side to the high pressure side so that when the available wind resource is weak, it is possible still to recover energy from storage (mode E). The presence of this compressor enables the system to draw net power from the grid and put energy into storage. Figure 1 shows the secondary compressor being driven by a separate motor connected to the grid but in a real implementation, it is likely that this machine would be coupled mechanically to the shaft of the main expander-generator set via a clutch to minimise power conversion losses.

Figure 2 shows that the gas on the high pressure side of the closed circuit can take any one of three distinct paths. In mode A, the gas bypasses the high temperature heat exchanger unit (HT HXU) and travels directly from the compressor exhaust to the expander inlet. In modes B and D, the gas flows through the HT HXU in a direction consistent with putting heat into the high temperature thermal store. Conversely, in modes C and E, the gas flows through the HT HXU in the opposite direction – consistent with extracting heat from the high-temperature thermal store. A similar three-way choice of paths is available on the low pressure side of the closed gas circuit and a separate illustration is not necessary. The two thermal stores in the system are ambient-pressure packed-bed thermocline-type thermal energy stores using solid particles of rock (or glass) to store heat/cold. Heat is exchanged between the high temperature thermal store and the closed gas circuit through the HT HXU using air as a heat transfer fluid to carry heat between the HT HXU and the thermal store. A similar arrangement is applied for the low temperature heat exchanger unit

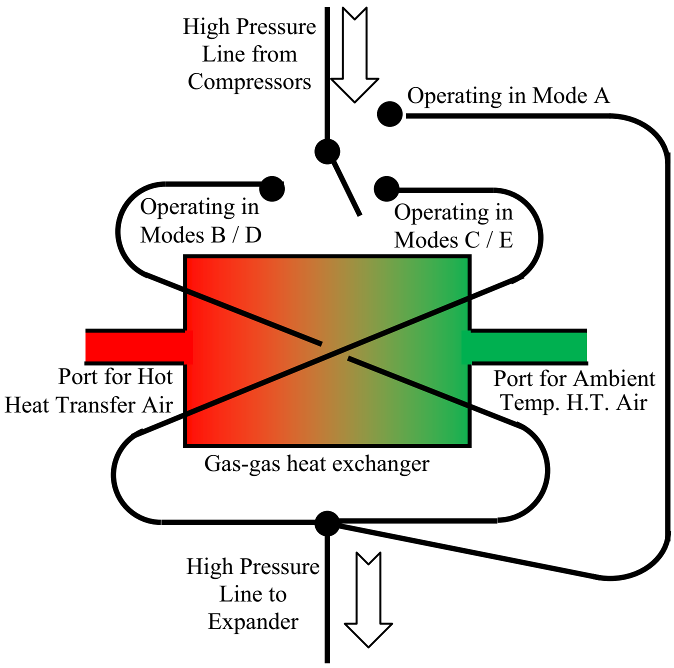

Figure 2. Three alternative paths through/past the HT HXU

(LT HXU) except that the air (or other gas) used as a heat transfer fluid on the cold side must not contain any components that may freeze.

Exergy is the ability to extract work (or electrical energy) from a system by allowing that system to come back into equilibrium with its environment. It has the same units as energy (J). Both mechanical work and electrical energy are pure exergy. Exergy is helpful in understanding how much heat (or coolth) we must move in order to store each 1J of electrical energy. Figures 3–7 provide simple illustrations of the five system operating modes showing how exergy is moved in each case. One further operating mode is conceivable – where energy is drawn into the secondary compressor, passed via the closed gas circuit to the expander and recovered through the expander-generator set with no heat/coolth being either taken from or inserted into storage. Because this operating mode serves no purpose other than possibly to verify the operation of system components, it is not discussed further.

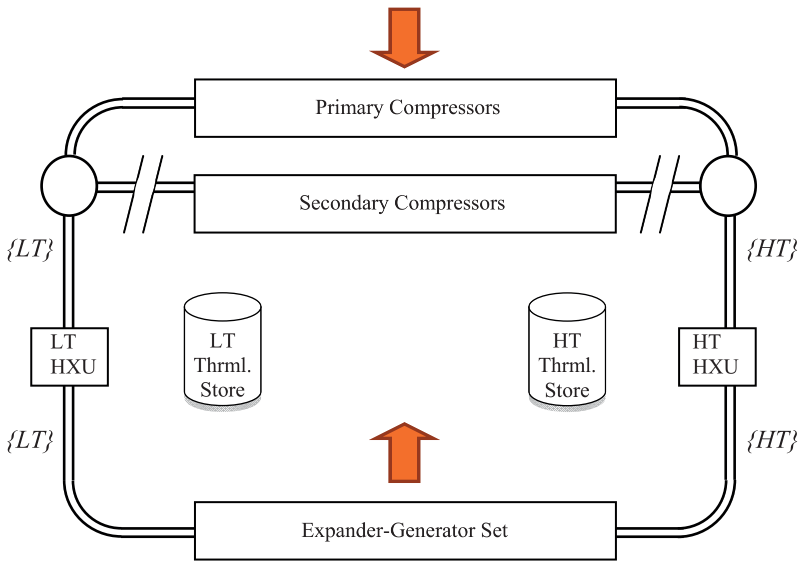

Figure 3. Exergy flows in operating mode A

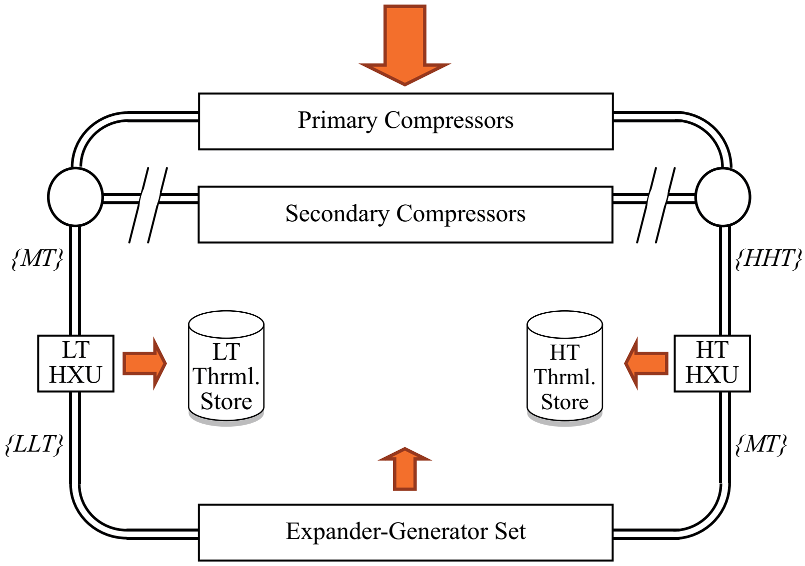

Figure 4. Exergy flows in operating mode B

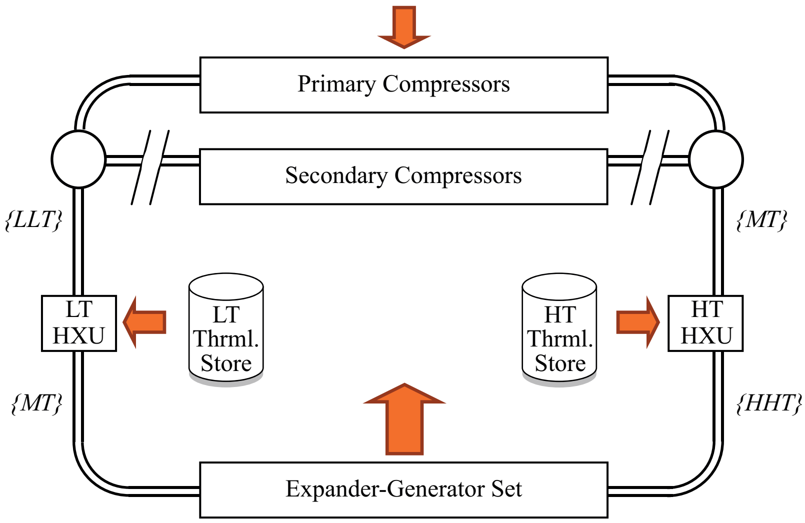

Figure 5. Exergy flows in operating mode C

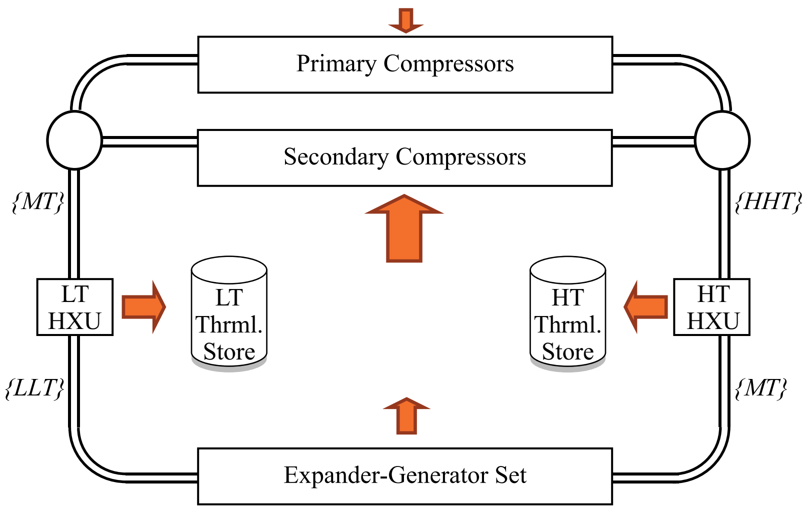

Figure 6. Exergy flows in operating mode D

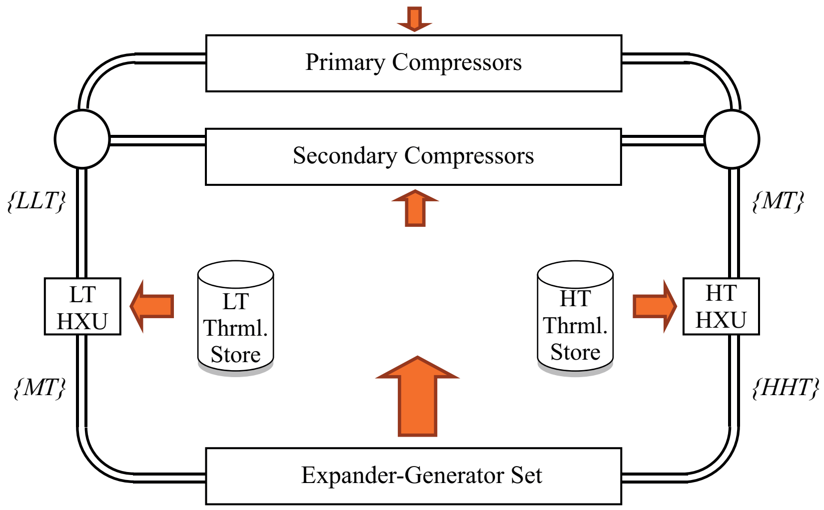

Figure 7. Exergy flows in operating mode E

The incremental exergy, ΔB, added into a system when a quantity of heat, ΔQ, enters that system is given by:

$$\Delta B = \Delta Q \times \left(1 - T_0 / T_1\right) \tag{6}$$

where T_0 and T_1 represent ambient temperature and the temperature at which the heat (or coolth) is added. Note that ΔQ is negative when heat is removed (i.e. when coolth is added) and thus removing heat from a system at temperatures below ambient ($T_1 < T_0$) accounts for an addition of exergy into that system. Table 2 below shows the exergy stored when 1000J of heat (or coolth) is stored at various different temperatures. Temperatures are reported on the absolute (Kelvin) scale and ambient temperature, T_0, is taken as 290K in this table.

All elements of the transmission system of interest will be as well insulated as possible from ambient since any natural heat transfer between the system and its surroundings is undesirable. However, it is inevitable that some heat transfer will take place. In any one of the system operating modes, parts of the closed gas circuit will have temperatures much higher than ambient and other parts will have temperatures well below ambient. For analysis purposes, we define a system reference temperature, T_R, as the geometric mean of the highest and lowest gas temperatures in the system at any one time. Taking account of small proportions of mechanical loss in compressors, the expander and the generator, T_R will remain slightly above ambient temperature. Variations in this temperature will be very small even as the system changes between operating modes. For the present, we estimate:

$$T_R \equiv 350K \tag{7}$$

Table 2. Exergy stored with 1000J of heat/coolth at different temperatures

Exergy added as heat			Exergy added as coolth		
T_1	ΔQ	ΔB	T_1	ΔQ	ΔB
900 K	1000 J	677.8 J	100 K	–1000 J	1900.0 J
800 K	1000 J	637.5 J	120 K	–1000 J	1416.7 J
700 K	1000 J	585.7 J	145 K	–1000 J	1000.0 J
600 K	1000 J	516.7 J	170 K	–1000 J	705.8 J
500 K	1000 J	420.0 J	200 K	–1000 J	450.0 J
400 K	1000 J	275.0 J	230 K	–1000 J	260.9 J
350 K	1000 J	171.4 J	260 K	–1000 J	115.4 J
300 K	1000 J	33.3 J	280 K	–1000 J	35.7 J
290 K	±1000 J	0 J			

In all operating modes other than mode A, two sections of the closed gas circuit will contain gas at temperature T_R (approximately). For brevity in the remainder of this section, we assume ideal behaviour of the system. Later sections examine the effects of irreversibility of various sorts. In all subsequent workings within this paper, we assume that the working gas is diatomic and has $\gamma = 1.4$. There is an interesting contrast here with the choice of another very carefully-considered energy storage system [2]. In that system, exergy is also stored through the storage of both heat and coolth in packed-bed thermocline-type stores and there is a closed circuit of gas. However, in that case, the thermal storage medium is contained within the gas circuit and thus the system cost is sensitive to the highest pressure in the system. This motivates the use of the monatomic gas Argon for which $\gamma = 1.6$ because this enables relatively high temperature swings for relatively small pressure ratios. The present system stores heat and coolth outside of the closed gas circuit and its cost is relatively insensitive to the upper pressure. Later, we will encounter a very sound reason for the selection of a particular diatomic gas, hydrogen, as the working fluid.

The operating modes of the system are described under the assumption that the pressure ratio (p_2/p_1) is 25. Then, given $\gamma = 1.4$, the ratio between temperatures emerges very close to 2.5. In operating mode A, gas in the low pressure side of the closed circuit will be at 221K ($\cong 350/\sqrt{2.5}$) and gas in the high pressure side of the gas circuit will be at 554K ($\cong 350 \times \sqrt{2.5}$). Exergy inserted by primary compressors in the wind turbines is withdrawn again by the expander. There are no interactions with the thermal stores in this mode.

In operating modes B and D, gas enters both the compressor and the expander at 350K. The gas leaves the compressor at 878K ($\cong 350 \times 2.5$) before giving up 528K in the HT HXU (see Figure 2). The gas leaves the expander at 140K ($\cong 350/2.5$) before gaining 210K in the LT HXU. Because the temperature of gas exhausted from the compressor is 2.5 times greater than that entering the expander, the mechanical power recovered from the expander is lower than that injected into the compressor by a factor of 2.5. Thus, if 20MW of power is injected into the set of all compressors, only 8MW will emerge from the expander.

In modes B and D the (negative) temperature swing experienced by the gas as it passes through the HT HXU is 2.5 times greater than the (positive) temperature swing experienced by the gas as it passes through the LT HXU. The specific heat capacity of a gas at constant pressure, c_p, is generally dependent on temperature but the variation is relatively small over the temperature ranges of interest here. Assume for the present that c_p is independent of temperature. Then it follows from the comparison of the thermal swings that for every 2.5 units of heat going into the hot thermal store, we would have 1 unit of heat being drawn from the cold thermal store. From an energy balance around the closed gas circuit, we can deduce that with 20MW of mechanical power entering through the compressor and 8MW of mechanical power leaving via the expander, the net incoming 12MW of mechanical power must be offset by the 1.5 “units of heat” leaving the system. In other words, the heat being pumped into the hot store must be 20MW and the heat being drawn out of the cold store must be 8MW.

Similar reasoning is easily applied to operating modes C and E where energy is being recovered from store. Gas leaves both the compressor and the expander at 350K. The gas enters the compressor at 140K after being cooled in the LT HXU from 350K. The gas enters the expander at 878K after being heated in the HT HXU from 350K. Because the temperature of gas exhausted from the compressor is now 2.5 times lower than that entering the expander, the mechanical power recovered from the expander is greater than that injected into the compressor by a factor of 2.5. Thus, if 8MW of power is injected into the set of all compressors, all of 20MW will emerge from the expander.

Assuming as before that specific heat capacity c_p is independent of temperature, we conclude that the ratio between the positive heat now returning into the gas circuit from the hot store and the heat being transferred to the cold store is again 2.5. Then with 8MW of overall input compression power and 20MW of output mechanical power emerging from the expander, there must be 8MW of heat flowing back into the cold store and 20MW of heat flowing out of the hot store.

The above paragraphs establish the energy flows in an idealised system but one further step is needed to convert these numbers to exergy flows. Integrating equation (6) between the two temperatures, T_0 and T_2 and assuming again that the thermal mass of any one elemental quantity of the gas does not vary with temperature (so that $\Delta Q \equiv M\Delta T$ for some M), we obtain:

$$\left(B_2 - B_0\right) = Q_{02} \times \left(1 - \frac{T_0}{T_2 - T_0}\log\left(T_2/T_0\right)\right) \tag{8}$$

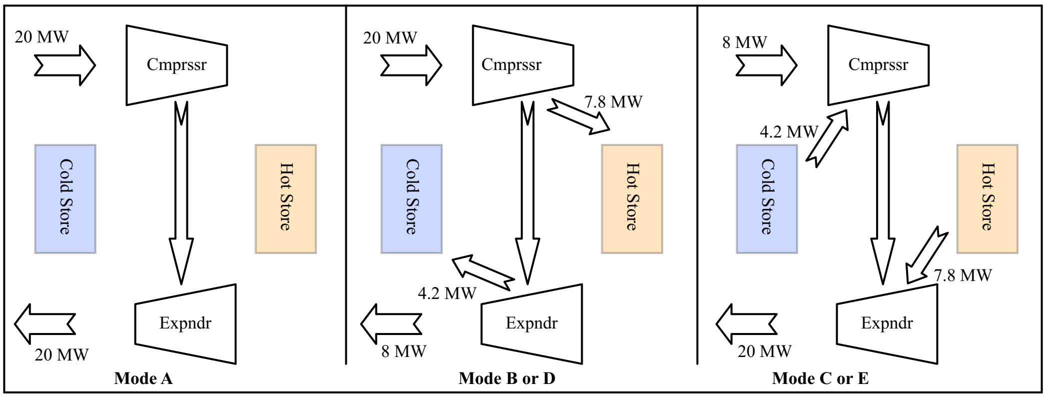

Figure 8. Summary of exergy flows for modes A, B/D and C/E

Here Q_{02} is the total heat injected into the system to change the temperature from T_0 to T_2 and $(B_2 - B_0)$ represents the corresponding total change in exergy. This equation shows us that when 20 MW of heat is flowing into (or out of) the hot store with upper temperature T_2 = 878K and with $T_0 = T_R$ = 350K, the exergy flow there is 7.8MW of exergy flowing into (or out of) the hot store. Similarly for the 8MW of heat flowing into/from the cold store with extreme temperature 140K, there is 4.2MW of exergy flowing into/from the cold store. Figure 8 depicts the flows of exergy for all operating modes.

With the pressures selected (and still using the simplifying assumption that c_p does not vary with temperature), we find that the hot store contains 65% of all of the exergy stored and the cold store contains 35% of it. However for a given duration of storage the total thermal masses required for the hot and cold stores are identical. For example, if a set of wind turbines with net output power of 20MW is to be equipped with 120MWh of storage (12MW × 10 hrs), the total thermal mass required for the hot store is (20MW × 10 hrs)/(528K) = 1.37×10^9 J/K and the total thermal mass of the cold store is (8MW × 10 hrs)/(210K) = 1.37×10^9 J/K.

It is worthwhile to put some scale to the above quantity of storage. The largest pumped-hydro power plant in the UK is Dinorwig pumped-hydro in Wales and this stores roughly 10GWh. The 120MWh of storage suggested above is 1.2% of the capacity of Dinorwig and would require around 1,600 tons of rock (or other material) with a specific heat of 850 J/kg K to form the hot store and the same quantity again to form the cold store. With ~60% packing factor in the packed beds, the mean density for the packed beds might be around 1,600 kg/m^3 and thus the internal volume of each thermal store might be 1000 m^3. A single tower with height 100 m and rms diameter of 3.6 m would accommodate this volume. During a complete charge/discharge cycle of Dinorwig, 6 million tons of water is moved.

4. POWER TRANSMISSION OVER KM-SCALE DISTANCES USING PRESSURISED GAS

The proposed system transmits power from the wind turbine rotors to the expander-generator set by driving a pressurised gas within the closed-circuit gas loop. One could envision each individual wind turbine having its own expander-generator set and its own thermal stores and heat exchanger units but (for onshore applications at least) the economics would be unlikely to stack up in this case. The alternative is to allocate only a primary compressor to each individual wind turbine and to locate all of the balance of plant in a central location. Figure 9 gives a schematic of such an arrangement with 7 wind turbines sharing one central provision. In that case, pipe runs in the order of 1km are to be expected since wind turbines are typically spaced apart by more than 7 times blade tip diameter [3] and blade tip diameters of >125 m are now not uncommon – though these tend to be offshore machines. Onshore machines are steadily edging up in size so we adopt 5MW as a realistic reference size of machine even for future onshore applications. In this section, we consider the viability of transmitting power over km-scale distances via compressed gas. The most fundamental concern is with pressure drop. This is calculated using the Darcy-Weisbach equation:

$$\Delta p = \tfrac{1}{2}\rho v^2 \times f_D \times \left(L/D_B\right) \tag{9}$$

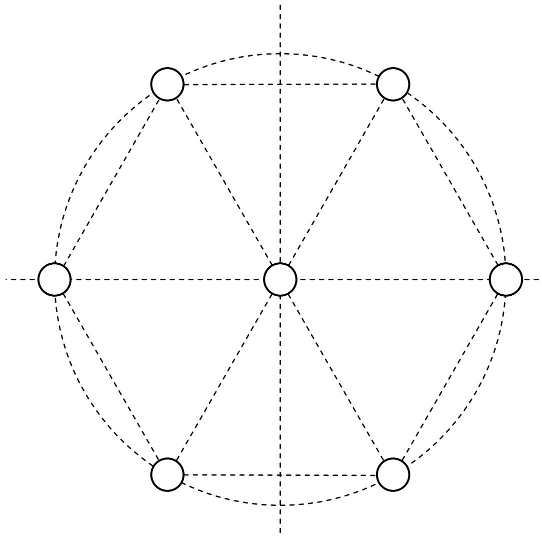

Figure 9. A single ***unit-cell*** of 7 wind turbines

In this, ρ represents the gas density, v represents the mean gas velocity, L denotes the length of the pipe and D_B represents its bore diameter. The friction factor, f_D, is determined based on Reynolds number for the gas flow and the relative surface roughness at the bore. In all cases of interest, we will have high values of Reynolds number ($Re > 1 \times 10^6$) and we will consider that relative roughness of around 1×10^{-4} is realistic. Then from a Moody diagram (e.g. [4] (page 271) or [5]), we find $f_D \approx 0.0135$. The exact value of f_D is not important – only that this factor does not vary strongly with Reynolds number.

Using reasoning already presented, it is already clear that the gas pressures on the low pressure side of the closed gas circuit should be several times greater than atmospheric pressure and that the pressure ratio between the two sides should be around 25. The operating temperatures in the closed gas circuit are fixed by the combination of the pressure ratio and a reference temperature, T_R which will be slightly above ambient. Up to now, we have not explored whether the gas pressures should perhaps be significantly higher – for example, might there be an advantage to having pressures of 100 bar and 2500 bar on the low and high pressure sides respectively? The first priority of this section is to examine this question.

The net power transmitted between a wind turbine and the central location in any one operating mode is the difference between the power being sent out in the high pressure side of the closed gas circuit and the power returning on the low pressure side. Both of these powers are proportional to pressure and volume flow rate. For either one, the power being transmitted is proportional to ($p \times v \times D_B{}^2$). Here p represents pressure, D_B represents the bore diameter and v represents the mean velocity of the gas within the pipe.

The power being lost in the transmission is proportional to ($\Delta p \times v \times D_B{}^2$). If our power transmission system was based on an incompressible fluid, it would be obvious that the fraction of power being lost in the transmission is simply ($\Delta p/p$). This is not true in the present case where a compressible fluid is being used but it is nevertheless true that the fraction of power being lost in transmission is directly proportional to ($\Delta p/p$) irrespective of the pressure p provided that the temperatures are fixed. With fixed temperatures, gas density is inversely proportional to pressure and hence we deduce from (9) that for a given power transmission length, L, and any given gas, the ratio (v^2/D_B) must be held constant to ensure that the fraction of power being lost in transmission is fixed. This indicates that

$$p \times D_B{}^{(3/2)} = constant \tag{10}$$

to transmit a given power over a fixed distance with a set allowable fractional loss. The material required in the pipe-work to contain the internal pressure is (approximately) proportional to ($p \times D_B^2$). From this

$$mass\ of\ pipe\ materrial \propto p^{-1/3} \tag{11}$$

Reducing the amount of material used in the pipes obviously reduces costs. It also reduces the thermal inertia of the pipework, the amount of insulation material required around the pipes to block thermal leakage and the mass of gas required in the system. The main argument against taking the pressures up to extremely high values probably lies in the selection of the expander. In the remainder of this, we set p_1 = 20 bar (2 MPa) and p_2 = 500 bar (50 MPa). Then the intake swept volume required for a 5MW compressor from Table 1 is only 1.69 m^3.

If a volume, V_1, of gas is initially at pressure p_1 and temperature T_1 and is expanded through a throttle so that its pressure falls to p_2, the exergy loss (relative to a reference temperature T_R) is given as

$$B_{loss,\Delta p} = p_1 V_1 \ln\left(p_1/p_2\right) \times \left(T_R/T_1\right) \tag{12}$$

This formula can be derived by considering the work which could have been extracted from that gas via isothermal expansion at temperature T_R from pressure p_1 to p_2. If the pipe bore diameters, length and roughness are known, we can apply equation (9) to determine the pressure drop and hence estimate the exergy lost using (12). Here, we take the reverse approach – setting out what exergy loss is acceptable in each pipe and using this to calculate the acceptable pressure drop. As an initial starting point, we accept exergy losses of 0.2% of the total input power of one wind turbine driven compressor due to the pressure drops in either the high-pressure or low-pressure lines when the system is operating in mode A.

For one single wind turbine operating at a rated power of 5MW, the inlet and outlet volume flow rates of gas will be 473.5×10^{-3} m^3/s and 47.51×10^{-3} m^3/s respectively – assuming $\gamma = 1.4$ for the gas and that the inlet and outlet pressures are 20 bar and 500 bar respectively. In mode A, the compressor inlet temperature is 221K and its outlet temperature is 554K. Applying (12), we can now calculate acceptable pressure drops in the low pressure and high pressure lines respectively as …

$$\Delta p(\mathrm{HP}) = 332 kpa, \quad \Delta p(\mathrm{LP}) = 13.3 kpa \tag{13}$$

This reveals that pressure drops equivalent to 0.66% of the nominal pressure are acceptable on both the high pressure and low pressure sides of the closed gas circuit. The minimum pipe bores are determined by these maximum acceptable pressure drops.

Noting that it is volume flow rates (and not mass flow rates) that are fixed by the power, it is obvious from (9) that using a low-density gas will be helpful. Accepting that f_D is (close to) constant, then evidently the pressure drop for a given volume flow rate is proportional to density and this immediately points to the use of hydrogen as a very low density gas. One factor not considered in detail here is that hydrogen does leak at a finite rate from almost every vessel and must be replenished. It also diffuses into some materials and can have adverse effects on their properties. Noting that it is a very cheap gas and notwithstanding the shortcomings noted above, hydrogen is considered as the default working gas of choice.

Applying the Darcy-Weisbach equation (9) to numerous possible pipe diameters for the high pressure and low pressure sides of the gas circuit allows the calculation of associated pressure drops. Then it is straightforward to determine a suitable pipe bore to produce the acceptable pressure drops. Figures 10a and 10b show the results of pressure drop calculations in the low pressure and high pressure sides of the closed gas circuit respectively when the system is operating in mode A. From these, it is straightforward to determine the required pipe bores. These are set as 205.3 mm and 71.3 mm for the LP and HP sides of the closed gas circuit. The values of Reynold's numbers are 857×10^3 and 1329×10^3 respectively.

The external diameters of the pipes are chosen by understanding what nominal stresses will be present in the tube walls over their lives. Hoop stresses in cylindrical pressure-vessels are greater than axial stresses – by a factor in the order of ≈2. Hence we focus mainly on the hoop stress. The low-pressure pipe will be a relatively thin-walled cylinder and for such cylinders,

$$\sigma_{hoop} \cong p_i D_i / 2t. \tag{14}$$

The low pressure pipe will never be subjected to high temperatures and this pipe will naturally tend to tolerate higher stress levels. Coupled with the fact that the pressure contained is relatively

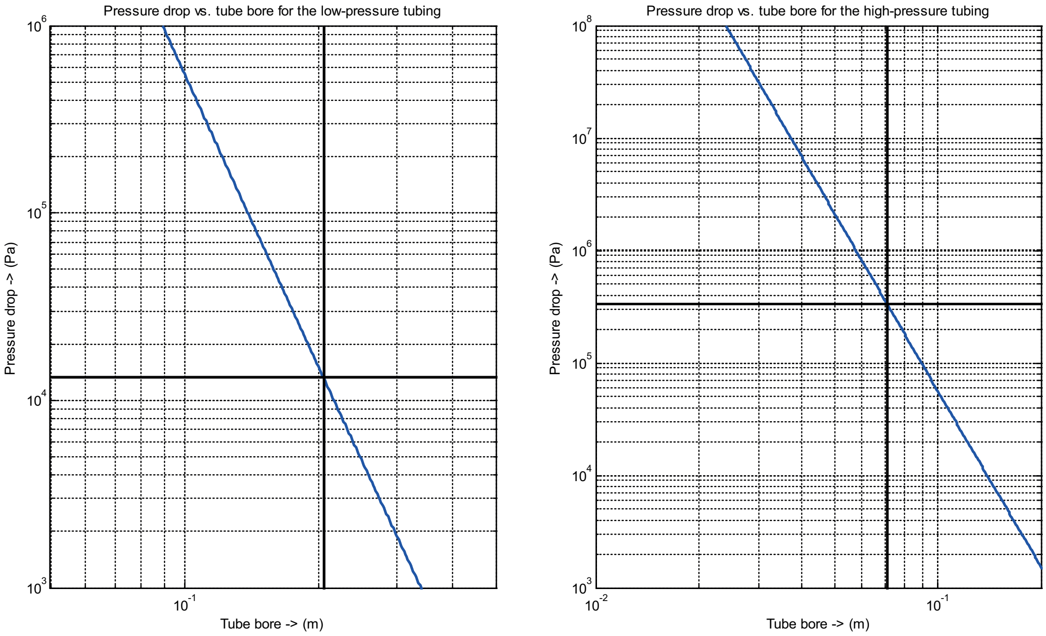

Figure 10. Pressure drop (Pa) vs. tube bore diameter for the LP and HP tubes respectively

low, this means that the minimum wall thickness of the low pressure pipe can be as low as 2 mm and the hoop stress will still fall below 103 MPa. A thin-walled pipe theory is clearly acceptable for this calculation.

The external diameter of the high-pressure pipe initially appears to require more consideration – since this pipe will have a wall thickness in the same order as the inner radius. One factor that simplifies the situation again is that the temperature of this pipe will rise considerably above ambient so that the material will become ductile and therefore hoop stresses will tend to equalise through the wall. In that case, equation (14) remains valid even for quite thick pipes. Applying this shows that with a wall thickness of 26 mm on the high pressure pipes, the mean hoop stress will be below 68.5 MPa. Obviously, because the high pressure pipes have to deal with high temperatures, the allowable hoop stress in these pipes is lower than that for the low pressure pipes. Table 3 summarises the pipe details including total mass of steel required for the low pressure and high pressure pipes for each individual 5MW wind turbine.

Although the total tonnage of steel for the pipework (72 tons per 5MW turbine) is obviously not trivial, it is very small compared with the tonnage of steel which would be used anyway in the construction of the wind turbine. Evidently the investment required for the pipework is not prohibitive. Note also that in view of equation (11), the material requirement could be reduced by choosing to operate at higher pressures.

The pipe dimensions above have been chosen based on 0.2% exergy losses in both the high pressure and low pressure sides of the gas circuit with the system operating in mode A at full load. The volume flow rates into and out of the compressor are the same with the system in modes B and C at full load but gas densities in mode B are 1.6 times lower than for Mode A and those in mode C are 1.6 times higher than the densities for mode A. Considering that the system will operate for a substantial proportion of the time with fractional load on the wind turbines (especially for mode C)

Table 3. Details of pipework for the LP and HP sides of the gas circuit (for each 5MW wind turbine)

Tube	Bore diam. (mm)	Wall thickness (mm)	Mean hoop stress (MPa)	Total steel mass over 700 m (kg)
Low Pressure	205.3	2	103	10,158
High Pressure	71.3	26	68.5	61,967

and that losses will be proportional to the square of the load fraction, the overall losses with this pipework are predicted to be <0.4% of the rated turbine power (5MW).

5. THE PRIMARY COMPRESSOR

The requirements on the compressor in question are quite singular in that:

- Extremely high efficiency is needed (>95% is important)
- The compressor is intrinsically a low-speed machine – ~15 rpm rated speed for a 5MW wind turbine
- The compressor must achieve near-adiabatic compression
- Inlet gas temperatures for the compressor vary from 140K to 350K
- The outlet gas temperatures for the compressor vary from 350K to 878K
- The compressor inlet pressure is already much higher than atmospheric pressure
- The compressor must be capable of operating at part load with good efficiency.

The high efficiency is essential in order that the straight-through conversion efficiency from rotor motion to electrical output at the generator terminals is at least comparable to the corresponding straight-through efficiency of conventional wind turbines. The arduous requirements for this machine may raise doubts as to its practicality. This section attempts to allay such doubts.

The very slow speed of the machine indicates immediately that a positive displacement machine is needed. The combination of slow speed with high isentropic efficiency indicates that a multi-stage machine is necessary. Normally, a designer of a near-adiabatic compressor would be content to use a single stage of compression and he would rely on the fast passage of gas through the compressor to limit the possible heat transfer. However, with slow throughput of gas, heat transfer between the gas and the machine becomes a much more serious concern. By performing the compression in several different stages thermally isolated from each other, the temperature rise in any one stage is not very high and the heat exchange between the gas and the compressor stage will be limited.

Figure 11 shows one proposed outline concept for this primary compressor as described in [6]. It shows five stages of compression driven from one common shaft. The gas path between the stages

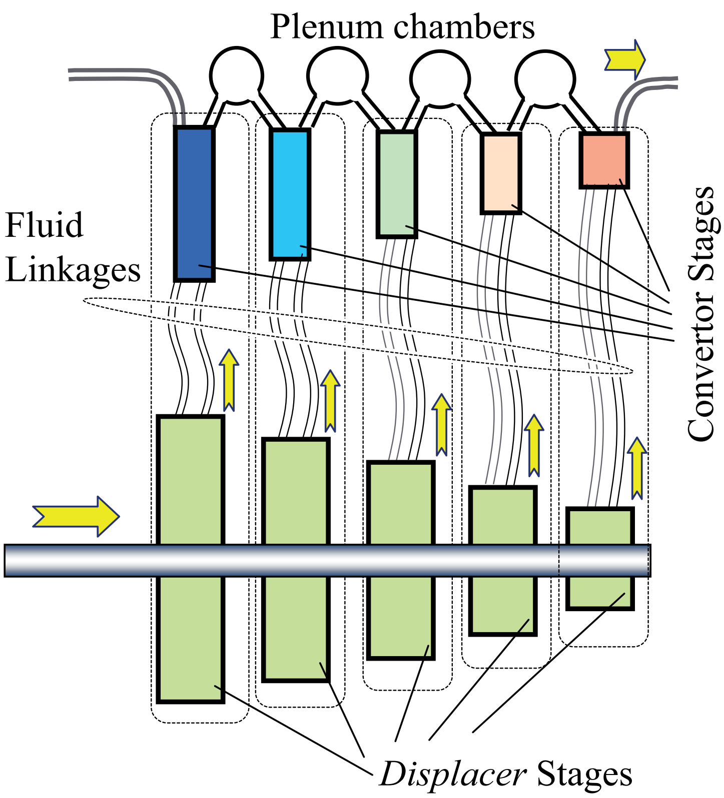

Figure 11. Compressor as a multi-stage machine

includes a plenum chamber between each consecutive pair of stages. Each stage of the compressor comprises both a *displacer unit* and a *convertor unit*. All of the displacer units are connected to a common power input shaft. A set of tubes forms a fluid linkage between the displacer unit in each stage and the corresponding converter unit. The arrows in Figure 11 indicate the flow of power. Figure 12 provides more detail on how this arrangement might look for a single stage of compression. In this figure, the displacer unit comprises a radial piston machine with an eccentric cam mounted directly on the input shaft and a cam ring following the cam. Anti-rotation features (not shown) prevent the cam ring from rotating but allow its centre to orbit. Connection rods between the cam ring and pistons in the machine cause each piston to alternate backward and forward relative to the cylinder enclosing it. As any one piston moves, the fluid in the cylinder above that piston undergoes corresponding movement through the fluid linkage and causes a liquid piston in one compression cylinder of the convertor unit to move.

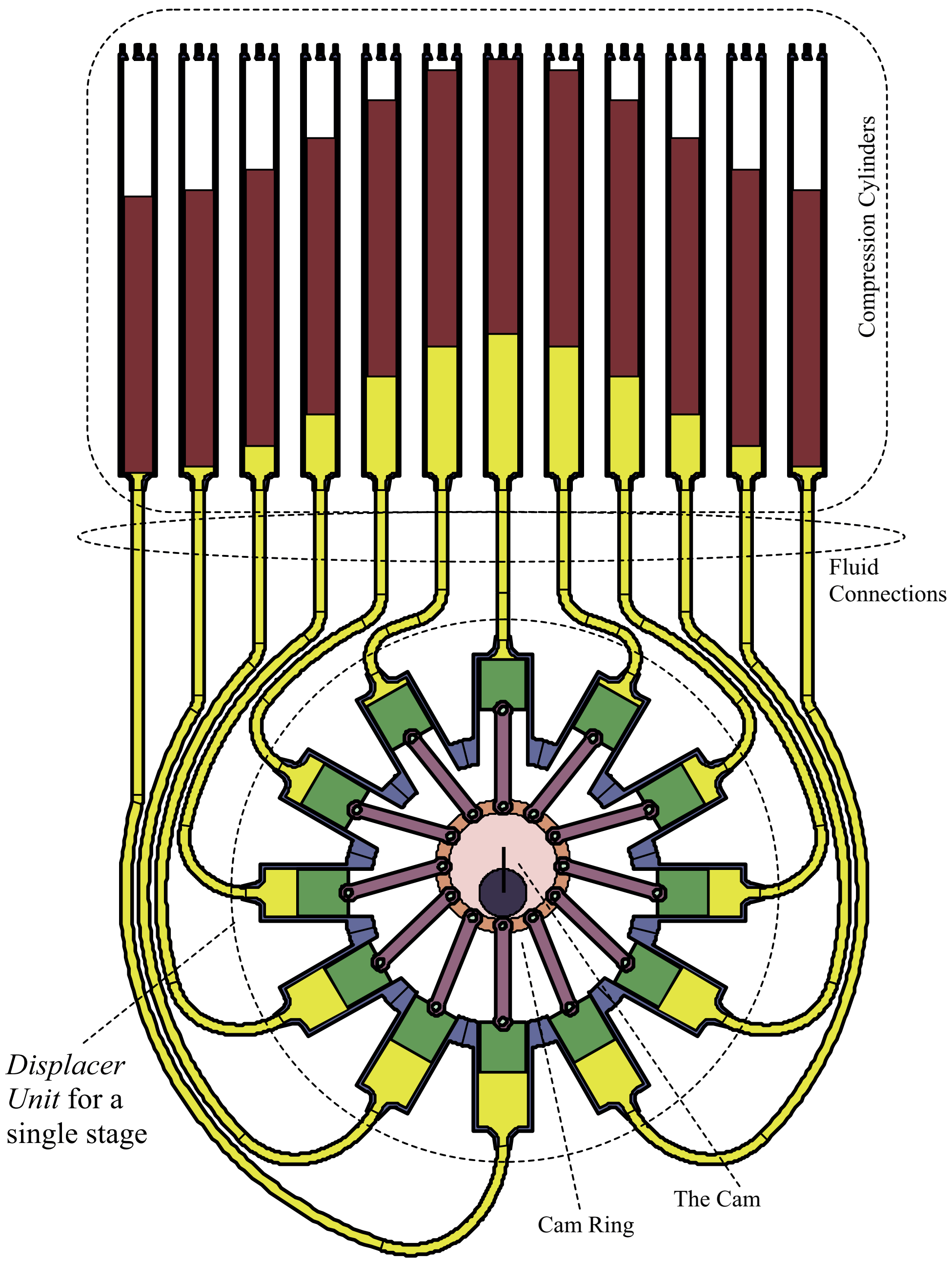

Figure 12. One single stage of the compressor

One important feature of the displacer unit design is omitted from Figure 12 for the sake of clarity. The volume inboard of the pistons in the displacer unit is filled with the same fluid that forms the fluid linkages and remains at a pressure approximately equal to the average of the inlet and outlet gas pressures for that stage. This greatly reduces leakage losses compared to what they might otherwise have been. A critically important element of the design of the convertor unit is the presence of *spacer pieces* floating atop the liquid pistons. The liquid pistons ensure that leakage of the gas is effectively zero and the spacer pieces provide a thermal barrier between the upper reaches of the cylinders comprising the convertor unit and the liquid forming the fluid linkage between the displacer unit and the convertor unit. The intention for the compressor design is that the liquid comprising the fluid linkages remains at a relatively constant temperature around 350K. The upper ends of the compression tubes in the convertor unit experience temperature extremes. In mode B, the gas leaving the final stage of compression will be ~878K and gas entering that stage would have temperature ~731K so the walls of the compression tubes would reach close to 804K. In mode C, the gas entering the first stage of compression will be ~140K and gas leaving that stage would have temperature ~168K so the walls of the compression tubes in this first stage would be expected to remain around ~154K.

The ability of the primary compressor to operate at speeds lower than rated speed is obvious. Nothing about the design of the compressor depends on speed. Despite being a positive displacement machine, this compressor can also operate at part load - independent of speed. This is fundamentally important for a wind turbine transmission so that energy yield from the rotor can be optimised. The part load capability demands that the valves at the tops of the compression cylinders are actively controlled. Then, any individual compression cylinder will draw no power from the compressor if the input valve to the cylinder remains open and the output valve is closed. Thus, any one stage of the compressor can be operating with 0 / 2 / 3 / 4 / 6 / 8 / 9 / 10 / 12 cylinders compressing and still remain balanced. Moreover, the plenum chambers shown in Figure 11 enable different stages of the compressor to operate at different load fractions. Since the starting point for the compressor design is that all stages of compression will draw approximately equal mechanical power, it is clear that there is very comprehensive control over total load. The expansion chambers shown at the bottom of Figure 1 provide a very small energy storage capacity capable of allowing output and input powers to the closed gas circuit to differ temporarily even within a single operating mode.

Three aspects of this compressor warrant quantitative discussion. The first aspect is size. If the machine has an intake swept volume of 1.69 m^3 and if there are 12 pistons in the inlet stage, then the swept volume of each of the cylinders in the inlet stage must be 0.14 m^3. If the stroke of each piston is identical to its diameter, then a diameter of 0.563 m is indicated. Scaling directly from Figure 12 would suggest that the diameter to the outside of the cylinders of the displacer unit is ~5 m and a more compact arrangement is clearly possible. The simplest arrangement for the convertor unit might suggest an additional ~5 m of height above the displacer unit. More compact arrangments are possible here also but even without these, the (5 m × 10 m) frontal area is not a serious concern compared with sizes for direct-drive generators. For comparison, the ENERCON E-126 machine has similar rated torque and uses a generator with 12m diameter stator [7]. Note, also, that if the first stage is implemented as two parallel stages feeding the same plenum chamber, all of the linear dimensions reduce by the factor 1.26. In this case, the second stage might well be mechanically identical to one of the two parallel stages for the first stage.

The second aspect of interest about the compressor is the leakage loss past the pistons. The volume flow rate of a liquid with viscosity μ (Pa.s), through a gap of height h, width W and length L (in the direction of the pressure drop) is Poiseuille flow (see [8]) and given by

$$Q = \frac{h^3 \Delta p W}{12 \mu L} \tag{15}$$

The viscosity of the oil used for the fluidic linkage might be 5×10^{-3} Pa.s. Allowing a generous radial clearance of h = 100 μm between piston and cylinder wall, taking the maximum pressure difference across one piston in the first stage as 1.5 MPa (the total pressure difference across the first compression stage is 3 MPa) and taking obvious values for both W and L, we discover a peak leakage rate across one piston in the first stage to be 77×10^{-6} m^3/s. The corresponding peak power loss is 115 W for one cylinder and the total power loss over the 12 cylinders in the stage would be around 692W. If losses in the other stages are similar, we anticipate a total power loss due to leakage in the compressor of <3 kW. This is clearly negligible. Calculations of the power loss associated with shearing of the oil as the pistons move relative to the cylinders are much smaller – peaking at around 12W per piston.

The final aspect of interest about the compressor is the rate of exergy loss due to thermal conduction vertically along the walls of the compression cylinders in the convertor unit. Suppose that the compression cylinders of the inlet stage have inner diameter 0.3 m and outer diameter 0.5 m. The stroke length of the liquid piston would then be 2 m and the spacer itself would be slightly longer than 2 m so that no part of the cylinder wall that is wetted by the liquid is also cooled by the intake gas. We suppose that (in mode C) the upper end of the cylinder is at 154K – around 200K cooler than the liquid. If the cylinder walls are made from a ceramic with thermal conductivity 2W/mK, the heat that would flow through the cylinder is calculated to be 25 W. Some more would flow through the spacer but since this will have a much lower effective conductivity, this is negligible. Evidently, this exergy loss is negligible.

There are many other details of the compressor to be considered and it is beyond the scope of this paper to explore all of them. It is not claimed here that the design outlined here is the optimal design or that all issues of this design have been resolved. However, the calculations above indicate that it is possible to build a compressor with the attributes required and that it is feasible to fit such a compressor into the nacelle of a large modern wind turbine. The authors believe it is realistic to achieve isentropic efficiencies of 98%.

The expander and the secondary compressor may each be either axial-flow machines or positive displacement machines such as screw expanders/compressors. In both cases, the preference is that these machines would be driven by synchronous electrical machines coupled directly to the grid. The actual designs may be quite similar to existing machine designs except that the casings must be much thicker and heavier than the cases of industry-standard machines today. It is not unusual for carefully designed axial-flow expanders to achieve >90% isentropic efficiency. Indeed, the expander efficiency in the electrical energy storage system currently being pioneered by ISENTROPIC Ltd. in the UK (and building upon [2]) must clearly be significantly better than 90% since turnaround efficiencies of >72% are claimed to be demonstrated and each 3J of exergy placed into storage requires approximately 5J of work to be done in a compressor and approximately 2J of work to be extracted from an expander. In discharging the system, the machines swap roles so both machines serve as both compressor and expander. If the isentropic efficiencies of the compressor/expander machines of that system were both 94% and there were no other exergy losses anywhere, the total losses would be around ((14/3) × 6%) = 28%.

Electrical machines of a size of several tens of MW often have efficiencies above 98%. Hence, we might reasonably expect power transmission losses (excluding flow losses and losses due to heat transfer) to be in the order of 14%. The previous section determined pipe diameters such that the losses due to pressure drops connected with the flow would be less than 0.5%. Exergy losses associated with undesirable heat transfer into the pipes are considered in the following section and it is shown that with realistic measures for thermal insualtion, these losses are below 1%.

6. EXERGY LOSSES ASSOCIATED WITH HEAT TRANSFER

The power transmission system relies on transferring heat effectively where heat transfer is required (into and out of the thermal stores) and on minimising unintentional heat transfer. Exergy losses due to imperfect or unintended heat transfer occur in three classes:

(1) Imperfect heat transfer between the closed-circuit gas and the hot or cold stores
(2) Heat leakage out from the hot store or into the cold store
(3) Heat leakage through the walls of pipes leading to the individual wind turbines.

We examine these in turn – applying exergy analysis to shed some light on the approximate magnitudes of exergy losses.

Imperfect heat transfer. With a gas pressure ratio of 25:1 and a reference temperature T_R = 350K, the hot-stores should be at 528K above T_R. A two-stage heat-transfer process is required to move heat between the closed-circuit gas and the thermal storage medium. The heat transfer between the closed-circuit gas and the thermal storage medium uses air as a heat-transfer fluid and the gas-to-gas heat transfer is the main thermal bottleneck. Note that the mass flow rate of the air used as a heat-transfer fluid can be controlled so that there is a perfect match of ($m \times c_p$) at all times on both sides of the heat exchanger. Then heat exchanger *effectiveness* is measured as the ratio of the temperature differences across both sides of the heat-exchanger. Values of *effectiveness* of 95% are not uncommon and effectiveness values arbitrarily close to 100% can be achieved – though heat-exchanger cost rises steeply as the losses reduce. We will take 95% as a representative achievable value of heat exchanger effectiveness.

For the hot-store, heat from gas emerging originally from the compressor at 878K would be transmitted into store as heat at (878 – 0.05 × 528) = 852K. That heat would be recovered again at (852 – 0.05 × 502) = 827K. For a given quantity of heat energy, ΔQ, transferred into the hot store and out again, we can utilise equation (6) to compare the exergy which would have been been passed through under conditions of perfect heat transfer with that passed through taking into account the temperature differences and the loss of exergy due to poor heat transfer at the hot store is found to be 4.1%.

For the cold-store, coolth from gas emerging originally from the expander at 140K would be transmitted into store as coolth at (140 + 0.05 × 210) = 151K. That coolth would be recovered again at (151 + 0.05 × 299) = 166K. Using equation (6) again shows that 26.1% of the exergy passing through the cold store could be lost due to poor heat transfer.

Recall, from section 3, that 65% of all exergy stored is retained in the hot store and 35% is retained in the cold store. Thus the overall loss of exergy due to poor heat transfer is assessed as (0.65 × 4.05% + 0.35 × 26.1%) = 11.8%.

A strong message emerging here is that heat transfer for the cold-store side demands very careful attention to ensure that the exergy losses incurred are not excessive. Very probably, the spend required to deliver heat exhangers which are much better than 95% effectiveness is justified for the cold-side heat-exchange.

Exergy Loss through the walls of the thermal stores. When the hot store is completely filled, the mean wall temperature will be maximised and the rate of heat loss through the walls will be at a maximum. Section 3 noted that a total volume of 1000 m^3 for both the hot store and the cold store was sufficient to contain 120MWh of exergy. It is advantageous for these stores to be tall and slender to minimise effects of natural convection in smearing out the internal temperatures. If each store was implemented as a cylinder of height 100 m and diameter 3.6 m, the total surface area of each would be 1174 m^2. If each cylinder was covered to a depth of 1m with an insulation material (mineral wool) whose thermal conductivity is 0.05 W/mK, then the thermal conductance between is 57 W/K. To be pessimistic, we assume that heat transfer from the outside of the insulation is perfect. With this conductance, and a temperature difference of ~580K between ambient and the internal temperature, a heat flow of 34kW would be expected. This would correspond to an exergy loss of only 20kW. This exergy loss corresponds to losing 0.4% per day. Obviously, with thicker insulation, this loss could be reduced.

The calculation for the cold stores is similar. The temperature difference from ambient is ~160K so the leakage of heat into the cold store (if it had 1m thick mineral wool insulation) would be 9.1kW and the exergy loss would be 13.7kW. Again, this is a trivial rate of exergy loss compared with the quantity of exergy stored.

Exergy Loss through the walls of tubes leading to the wind turbines. As noted in section 4, it is normal for the spacing between turbines to be more than seven times the blade tip diameter in order that the wakes of upwind turbines do not overly compromise the achievable output from downwind turbines. With the proposed power transmission system, the power from each one of six of the seven turbines in one group travels radially inward by a distance of several hundred metres before being converted from the form of a compressed gas into electrical output power. The tubes communicating the gas in the circuit between the central location and each one of the satellite wind turbines have a finite external area and some exergy will be lost through that area. Here, we show that the proportion of exergy being lost by heat transfer through these tube walls may be <1% in operating modes B or C with relatively modest insulation arrangements. Then in operating mode A where the system is acting as a pure power-transmission system, the exergy losses are lower. As before, we take the pessimistic view that heat transfer is perfect at the outer and inner radii of the insulation.

The thermal conductance, K, of a thick cylindrical layer of insulation between radii R_i and R_o with length L is given by [9].

$$K = 2\pi kL/\log(R_o/R_i). \tag{16}$$

Table 4. Exergy losses in the long pipe runs to individual 5MW wind turbines

	Hot insuln. (R_i = 62 mm, R_o = 375 mm)	Cold insuln. (R_i = 105 mm, R_o = 375 mm)
Mode A	Internal Temperature = 554K Heat Leakage (out) = 37kW Exergy Loss = 14kW	Internal Temperature = 220K Heat Leakage (in) = 14kW Exergy Loss = 8kW
Mode B	Internal Temperature = 878K Heat Leakage (out) = 82kW Exergy Loss = 49kW	Internal Temperature = 350K Heat Leakage (in) = 12kW Exergy Loss = 0
Mode C	Internal Temperature = 350K Heat Leakage (out) = 8kW Exergy Loss = 0	Internal Temperature = 140K Heat Leakage (in) = 30kW Exergy Loss = 45kW

In this, k represents the thermal conductivity of the insulation material and we take k = 0.04 W/mK which is achievable with Perlite or several different mineral wools. As before, L represents the length of the power transmission and we set L = 1000 m here. The target total loss of exergy is 1% of 5MW or 50kW. In mode B, the main exergy loss is from the high pressure pipe at temperature 878K. For this temperature, 50kW of exergy loss corresponds to 83kW of heat loss. If the temperature difference to ambient is 590K, the maximum thermal conductance allowable is K = 142 W/K. From this, the ratio of radii R_o/R_i for the insulation on the hot pipe is calculated to be 5.9. Earlier, the outer diameter for the hot pipe was found to be 124 mm, so the outer diameter of the insulation on the hot pipes would then be 733 mm.

In mode C, the main exergy loss is from the low pressure pipe at 140K. Here 50kW of exergy loss coincides with 33kW of heat loss but the temperature difference driving this heat loss is lower – at 150K. This corresponds to K = 220 W/K. The ratio of radii R_o/R_i for the insulation on the cold pipe is calculated to be 3.13. With the outer diameter of the cold pipe being 210 mm, the outer diameter of the insulation is then 657 mm.

In the main power transmission mode of operation, mode A, the temperatures are less extreme but exergy losses occur from both the hot and cold pipes. Table 4 summarises the exergy losses from the long pipes for the three modes of operation where the outer diameter of the insulation has been rounded up to 750 mm.

The total volume of insulation used here for a group of 7 wind turbines (only 6 of which have long pipe runs) is 5020 m^3. If the insulation material is expanded Perlite, the mass of this would be around 500 tons – similar to the mass of steel employed in the pipework but much less expensive.

7. OTHER PRACTICALITIES

The main objective of this paper is to suggest that a particular overall scheme for integrating energy storage may indeed be feasible. Several practicalities of the implementation are not addressed in full detail because they are beyond the scope of this paper. These include the making-up of small leaks of gas from the closed gas circuit, maintaining constant pressures on the hot and cold sides of that circuit even between modes B and C where temperatures of most of the circuit volume rise by a factor of (5/2), operating the valve gear in the compressor, compensating for slow net leakage of oil between stages in the compressor, thermal isolation of stages of the compressor from each other, operation of the compressor in part load conditions and design of the valving arrangement for diverting the gas flow through the heat exchangers in different directions (or past the heat exchangers) as Figure 2 shows. In each case, it is fairly evident that multiple different approaches are possible and that there is a high probability that at least one of those approaches will be practicable.

The power transmission system proposed here is compatible with either horizontal-axis wind turbines or vertical-axis machines. In the former case, one significant practicality concerns how to achieve a rotatable coupling to accommodate yawing of the nacelle. More specifically, the hot and cold pipes in the stationary frame of the tower must be connected to corresponding hot and cold pipes in the rotatable frame of the nacelle. This is not a trivial concern. The electrical connections for modern horizontal-axis wind turbines do not use a rotating connection. Instead they allow the electrical cables to twist by several full revolutions and yaw control logic within the turbine acts to "unwind" the cables after that point. With a low-pressure pipe of outer diameter 210 mm extending from the foot of a 100 m tall turbine tower up to the nacelle, a twist of 180 degrees would cause shear strains of 3300×10^{-6} and corresponding shear stresses of 270 MPa. Clearly, this is not viable. Forming the low and high pressure pipes into helices could provide the capability for elastic twisting of one or two revolutions – but at the expense of lengthening the pipe runs.

A rotating seal set is required which simultaneously makes the connection for both high pressure and low pressure sides of the closed gas circuit. The seal set must keep hot and cold streams separate. A very limited amount of leakage between the two streams is acceptable. We can calculate the associated exergy loss easily and this is done below. The seal should have a virtually zero net leakage of system gas out to atmosphere. The seal set must handle the significant pressure loads attempting to separate it. It must also tolerate inlet (LP) gas temperatures between 140K and ambient and outlet (HP) temperatures varying between ambient and 878K – though all of the temperature changes will be quite gradual. It must not require substantial yaw torque to turn it. Most importantly, it should have an expected lifetime consistent with 30 years or more of wind turbine operation without replacement.

Wind turbines do not yaw very much in a lifetime. Large machines have a limit on yaw speed of 0.3 degrees/s [10] and at this speed it takes 20 minutes for a machine to yaw through a full revolution. Even if these machines were yawing all of the time at this speed, they would clock-up only 72 revolutions in a day. In reality, the yaw action takes place only when the mean wind direction is significantly displaced from the turbine axis in angle. A very pessimistic assessment is probably 3 revolutions per day on average. Then a lifetime of 30 years would require ≈30,000 revolutions.

A seal configuration based on super-flat annular Zirconia faces pressed against each other is envisioned. Figures 13a – 13c outline the nature of an adaptor which could bring together the hot and cold pipes into a pair of concentric annuli. Seals based on the contact of two super-flat Zirconia faces are already in common usage in the petrochemical industry and also in the common domestic tap (USA:*faucet*). With suitable provisions made for loading the faces together, the pre-loading forces would be offset almost completely by the pressure reaction of the seals so that the net reaction force at the contacting faces was very low.

A 5MW primary compressor operating between 20 bar and 500 bar (for a gas with $\gamma = 1.4$) will require an inlet volume flow rate of 0.474 m^3/s. With this compressor operating in mode (a), the inlet temperature would be 220K and the mass flow rate would therefore be 1.04 kg/s. Suppose that the annular faces separating the cold and hot streams have mean radius of 150 mm and radial depth of 20 mm and suppose that a constant gap of 1μm is sustained between these two faces. It is useful to calculate the flow rate of hydrogen through such an annular gap. By discretising the annulus into, say, 200 consecutive thin annuli of radial thickness 0.1 mm each, we can consider that the gas density is constant through each one and the pressure drop for a given thin annulus can be computed using a rearrangement of equation (15). After the gas has passed through any one elemental annulus, the pressure has fallen and the volume flow rate must be increased accordingly for the next annulus. Figure 14 shows the effect of this in the profile of pressure against radius for a given mass-flow rate of 0.43 g/s (0.042%). Exergy calculations show that this accounts for a loss of exergy of around 0.1% of the wind turbine output. Evidently, if we can find a way to keep the two surfaces of the rotatable seal very close (~1 μm), the leakage between hot and cold passages can be reduced to a negligible level.

Leakage between the low-pressure volume and the atmosphere can be managed by using a viscous liquid to form the seal and using very occasional (negligible-power) pumping to drive some of that liquid back to chambers at the inner parts of that seal.

The power transmission system of this paper would probably have a very different complexion for offshore machines. Transmission of hot or cold gas over long distances underwater between large offshore machines is very unlikely to be effective given the relatively strong possibility that seawater would leak into the insulation space. In view of this and with the turbines themselves being larger than onshore machines, the argument for having one expander-generator set and one pair of thermal stores on board each fixed or floating wind turbine support is strong. Floating wind turbine platforms are becoming ever more common now and one class of floating platform involves comprises platforms that yaw bodily upon the surface of the water [11]. If the primary compressor of the proposed power transmissions sytem was mounted in the nacelle of a wind turbine upon such

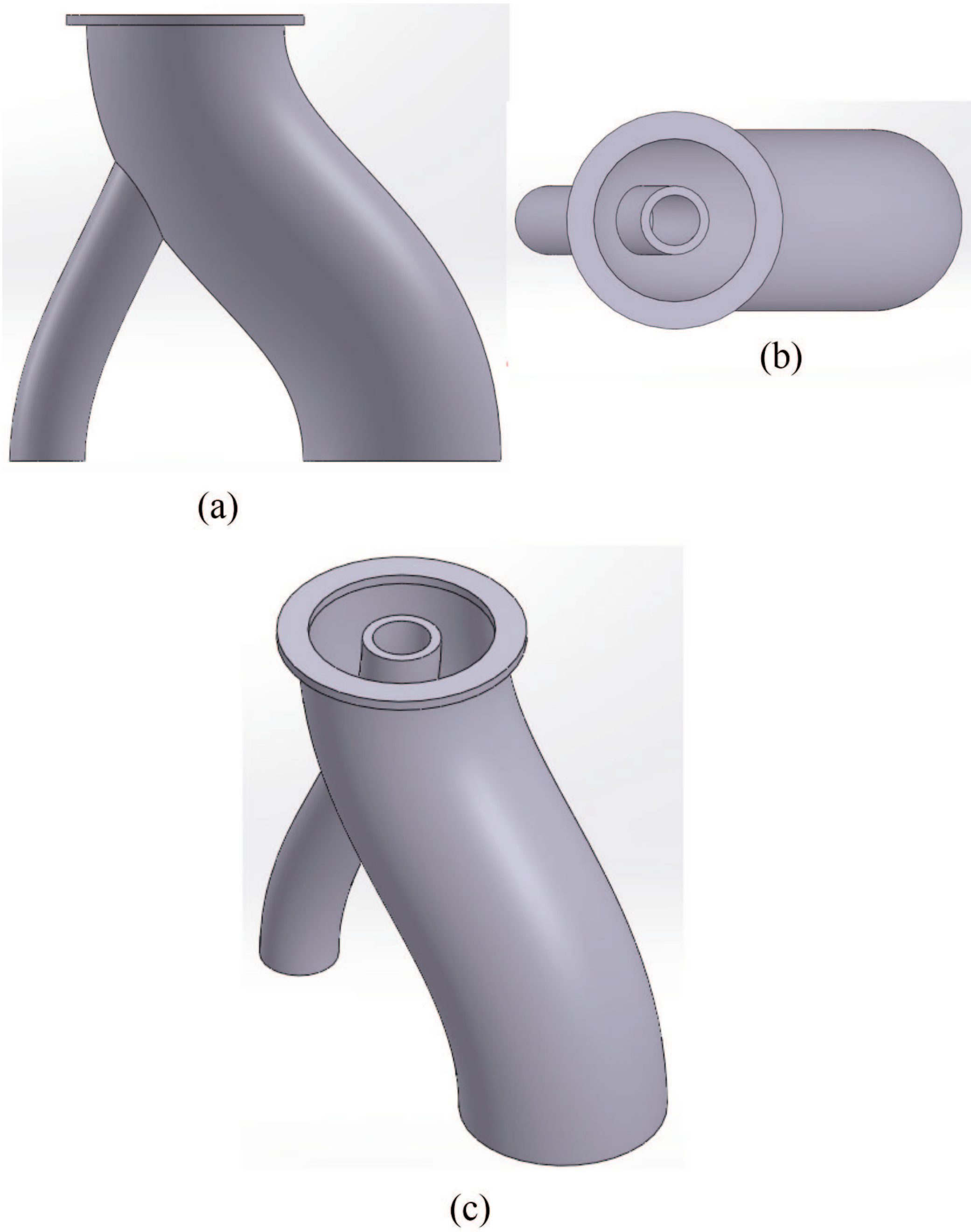

Figure 13. (a) An adaptor: front elevation; (b) An adaptor: plan view; (c)Oblique view of an adaptor for a rotatable coupling

a floating platform, there is no requirement for a rotating seal and the total length of the transmission pipes between primary compressor and location of the expander and thermal stores is very dramatically reduced.

One other practicality deserves mention here – thermal inertia of the system. A detailed study is needed to understand how to manage transitions between different operating modes of the system. However, it is possible to obtain ball-park estimates of what time-constants are involved. The main thermal inertia for any one wind turbine in the proposed system is associated with the pipework connecting that turbine to the central location. Earlier, the mass of steel required per 5MW turbine was assessed as 72 tons – 10 tons of steel for the low pressure pipe and 62 tons for the high pressure pipe. Taking the specific heat of steel to be ~500 J/kgK this indicates thermal masses of (5×10^6) J/K for the low pressure pipe and (32×10^6) J/K for the high pressure pipe.

Changing the operating mode from mode A to mode B would cause temperature changes of 81K and 204K in the low pressure and high pressure pipes respectively. These correspond to heat

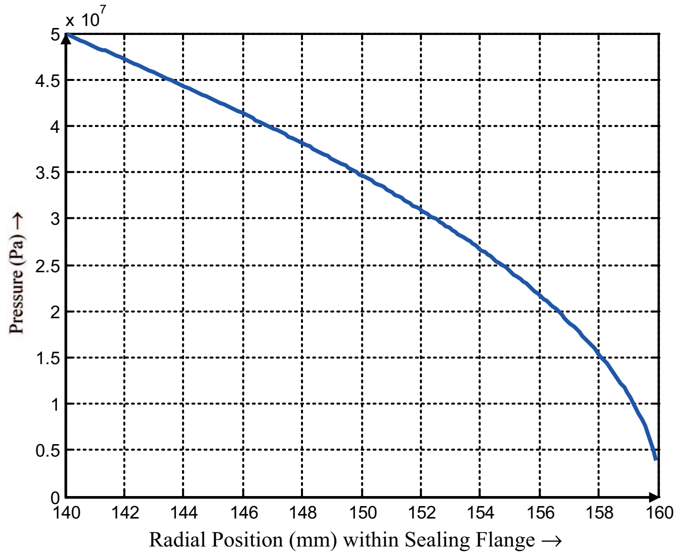

Figure 14. Pressure drop (Pa) vs. radius (mm) through a plain annular gap

movements of (405×10^6) J and ($6{,}530 \times 10^6$) J. In total, ~7GJ of heat is moved. The rated machine power is 5MW, simply changing mode will clearly require a time in the order of 30 minutes. It is important to note that remaining in mode A does not require that the wind should remain at or above rated wind speed. Any one of operating modes A-C can be operated at part load to suit conditions and there is no restriction on changing the load level in any one operating mode arbitrarily quickly. It is also important to note that the main effect of the thermal inertia is a delay in the actual implementation of a mode-change rather than a large loss of exergy – though clearly some exergy is lost in each mode change through irreversibility in the heat transfer. Work into the loss of exergy per instance of mode change is ongoing. The power-transmission system introduced here is evidently suited mainly for longer-term storage. As noted earlier, the expansion vessels indicated at the bottom of Figure 1 provide a capability for short term energy storage without the system changing between different operating modes.

8. THE COMMERCIAL CASE

At present, incentives for wind power provide the main source of financial return in most places and these incentives are based simply on the cumulative electrical energy delivered with no consideration for how this correlates with demand. In such an environment, a power transmission system with integrated energy storage capability will not seem appealing to a wind turbine manufacturer or operator since the new power transmission system will cause the system to cost more for the same net electrical output. Some rewards are in place for energy storage systems but currently these do not provide a very strong motivation for investment in the development of energy storage provisions. A small number of projects have been put underway to develop experience with energy storage. Although the present integrated system of power transmission and energy storage can be used as a standalone energy store (operating modes D and E), its performance is very ordinary in that context. It is most unlikely that any initiatives designed exclusively to help germinate new technologies in energy storage will provide a platform for an integrated storage technology.

The case for systems integrating energy storage with wind energy harvesting relies on both a strong demand for clean energy and a reward system being in place to reflect the full value of energy storage. In an ideal future, the incentives schemes for wind power would be re-fashioned such that the reward for generation from wind would become dependent on the demand for power at the time of delivery. In such an environment, parties intending to deliver wind farms would be minded to compare a conventional wind farm with standalone energy storage provisions with a wind farm having some quantity of integrated storage. To complete this comparison in detail would require an integration over time using some typical expected profiles of wind resource and demand. However, an approximate assessment may be made by estimating some overall system-level quantities.

Table 5. Comparison of cost components between two wind farm arrangements incorporating storage

	Conventional wind farm with separate (standalone) storage	Integrated power xmission & energy storage
Cost excluding storage costs	\$200M	\$240M
Marginal costs, energy storage capacity	\$72M (based on ~\$30/kWh)	\$30M (based on ~\$12.5/kWh)
Marginal costs, power into storage	\$56M (based on \$700/kW)	\$0M (already paid for)
Marginal costs, power from storage	\$175M (based on \$700/kW)	\$80M (most already paid for)
Cost of Energy Loss for energy passed through the storage over 20 years.	\$313M (based on \$100/MWh & turnaround efficiency = 70%)	\$157M (based on \$100/MWh & turnaround efficiency = 85%)
TOTAL Cost	**\$816M**	**\$507M**

The main parameters of a facility providing combined wind energy harvesting and energy storage are:

- rated power of the wind farm
- nominal energy storage capacity
- rated power associated with charging up the energy stores
- rated power associated with discharging the energy stores
- effective turnaround efficiency associated with passing energy through storage.

Of the above, the first four can be increased or decreased independently with several possible system designs and in each case, it is usually possible to associate a marginal cost with increasing the parameter. The number of variables of the system can be reduced by simply normalising the second, third and fourth parameters relative to the rated power of the wind farm.

Systems can then be compared by selecting certain ratios that are likely to be near-optimum for a particular context. An example might be:

- the wind farm will have a rated power of 200MW
- it is to be equipped with energy storage capacity equivalent to 12 hours of full rated capacity – i.e. 2.4 GWh
- the power rating for putting energy into storage should be 80MW (40% of wind farm rated power)
- the power rating for recovering energy from storage should be 250MW (125% of wind farm rated power).

The comparison between different options would be completed by estimating what fraction of energy would be expected to pass through storage with the above parameter set and what mean value was set on delivered output energy. In this case, the comparison might look as Table 5 indicates. In this, it is assumed that 25% of the total energy output of the wind farm over 20 years (10.5×10^6 MWh) will have passed through storage. This table uses only indicative values for costs. Its purpose is to emphasise that the direct cost component of the wind energy harvesting hardware is only a small fraction of total system costs over a lifetime and that even though the new transmission system may cause a slight increase in direct costs of a wind farm, it may result in a very substantial saving of money.

9. CONCLUSIONS

This paper has set out an analysis approach for wind generation systems in which a new power transmission format is adopted so that it can endow the system with the capability to store energy. A large part of the paper is devoted to arguing that the new power transmission format is indeed practicable. Different formats appear suitable depending on whether the wind generation system is onshore or offshore. Onshore, the optimum system uses a single expander-generator set and pair of thermal stores to serve a set of 7 wind turbines. Offshore, there are strong arguments for using floating platforms which can yaw above the sea surface and deploying one single expander-generator set and pair of thermal stores per floating platform – even if that platform supports only one rotor.

Two limitations are highlighted by the analysis: (a) changing between operating modes of the system appears to have a minmum time-constant in the order of one hour (for very large onshore wind turbines) and (b) a maximum of 60% of the energy being harvested by the wind at any one time can be put into storage with high efficiency (i.e. using mode B). Larger proportions of the energy being collected can be pushed into storage if the secondary compressor is also run.

Calculations show that with careful design attention, there is a good prospect that systems can be engineered having a straight-through transmission efficiency (from wind turbine rotor to terminals of the generators) of >85%. It is useful to compare this with values achieved by other power transmission systems (rotor power to electrical terminals). In [12], five different arrangements are considered and estimated figures are provided for total losses and total output electrical energy over one year. The values show total transmission efficiencies between 91% and 94%. The reference machine of [10] is given an efficiency of 94.4% (at full load). The hydraulic transmission system of ARTEMIS IP presented in [13] anticipates a peak efficiency (mechanical power from turbine rotor to mechanical power at the generator(s)) of around 93% but generator losses will drop this to perhaps 91%. An important point made in [13] is the large significance of good efficiency at part load as well as at full load. A major element of the design to achieve that in [13] is the provision of two generators and associated hydraulic motors. For the system described in this paper, the <1% of full load exergy losses due to thermal leakage in the pipework will not reduce with part load. Exergy losses in the compressor will reduce in (almost) direct proportion to load. Exergy losses due to pressure drops associated with flow reduce with the square of load. Good part load efficiency can certainly be achieved if the expansion facility is divided into two or more machines.

The effective turnaround efficiency (defined here as the ratio of electrical energy actually delivered from the system after energy has passed through storage to the energy that could have been extracted if energy had not passed through storage) is also in the order of 85%.

Exergy losses of several different types have been identified and quantified. The most notable of these is associated with the performance of the heat exchange unit for the cold stores. It is obviously worthwhile to lavish more engineering effort on this one component in particular to raise system performance generally.

The section on commercial arguments shows that if systems such as the one espoused in this paper are set alongside combinations of conventional wind farms with conventional energy storage provisions equivalent to several hours of energy storage at rated output power capacity, the new system format emerges very favourably in terms of overall costs.

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