Sea-plow: an artificial ocean upwelling device for carbon sequestration

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Discussion

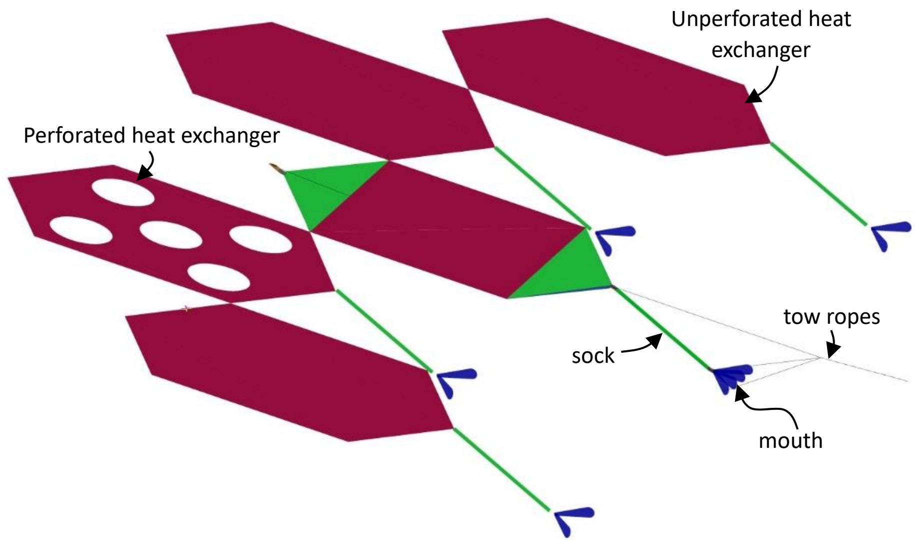
Sea-plow is an 'artificial ocean upweller' composed of a large tube of flexible low-density polyethylene (LDPE), aka 'garbage-bag' plastic, glued to fishnet and dragged through the ocean at 0.7 mph. This generates a maximum internal gauge pressure in the plastic of 2 kPa (0.3psi). This plastic tube is held in various shapes by internal plastic restraints. The mouth of sea-plow is located 600m below the surface, and water entrained at that depth is raised to a depth of 60m, above the ocean's thermocline, through a plastic tube supported internally by a truss composed of high-density polyethylene (HDPE). The water then enters a heat-exchanger section, composed of a flat plate with a thin internal channel, that is 1360m wide and 2000m long, but whose channel is only 0.57m high. This has the effect of bringing the cold (4 C) water entering sea-plow up to a temperature closer to that of the ocean surface (15 C). The water then enters a short tube and is deposited at the surface.

In artificial ocean upwelling, nutrient-laden water from deep in the ocean is brought to the surface to fertilize sun-drenched surface waters. If done properly, in temperate and polar oceans during local summer, this eventually results in the entrainment of atmospheric CO2 into ocean biomass, much of which sinks to the ocean bottom (Refs 1-7, 10). Climate/ocean modelling of artificial ocean upwelling has indicated that deep ocean water must be brought via heat exchange to a temperature nearer that of the ocean surface before being injected into the surface. Otherwise, unanticipated thermal effects would occur that counter the nutrient fertilization effect of upwelling (Ref 6). It is for that reason that sea-plow incorporates a heat-exchanger in its design.

Estimates suggest sea-plow can perform this function at a cost of $278/metric ton of CO2 sequestered. This includes the cost of towing the sea-plow, which is estimated to require 5.4MW of towing power (since towing power is proportional to the velocity cubed, this is the primary reason for the very low towing speed of 0.7 mph). A $100/ton carbon sequestration cost is considered the dividing line for economical sequestration, so perhaps with improvements made possible due to a learning curve, sea-plow may eventually be cost-effective (Refs 8, 9). Also, other benefits from fertilizing the surface ocean, such as potentially larger fisheries, could advantage sea-plow (Ref 10).

It was randomly chosen to design each sea-plow so that 40,000 of them operating in the ocean could sequester enough atmospheric CO2, in a year, to lower it by 1ppm (it is currently above 400ppm). The sequestration rate for deep ocean water brought to the surface from 600m down is about 0.12 moles C/m3 of upwelled water (Ref 5). This requires that each sea-plow entrain 1170 m^3/s of sea water from 600m depth and lift it to the surface. The annual cost of operating 40,000 sea-plows is estimated to be $2 trillion. This number of devices, placed edge to edge, would take up about the same area as Alaska, but would be distributed in high-latitude oceans like the North Pacific, North Atlantic, and the Southern Ocean. Sea-plow is in a preliminary stage of development so all of these are early estimates.

Figure 1: Sea-plow, composed of five units



The following discussion outlines the current mechanical and thermal design of sea-plow, followed by a discussion of the economic analysis. Figure 1 shows the current sea-plow design. It is composed of five identical units arranged in a checkerboard fashion so that the heat exchanger sections don't cover too much of the ocean surface at a time. It remains to be understood what effect the large area of the heat exchanger has on marine life. In particular, air-breathing mammals may be impeded by the heat exchanger held 60m below the surface. Further study will be needed to understand how great a threat this is to wildlife. Figure 1 also shows one of the sea-plow units that has a heat exchanger that is perforated with five holes. These holes would allow sea-life greater access to the ocean surface but would necessitate enlarging the heat exchanger from the standard design, so that its over-all heat-transfer area remains the same.

Figure 2: An individual sea-plow unit

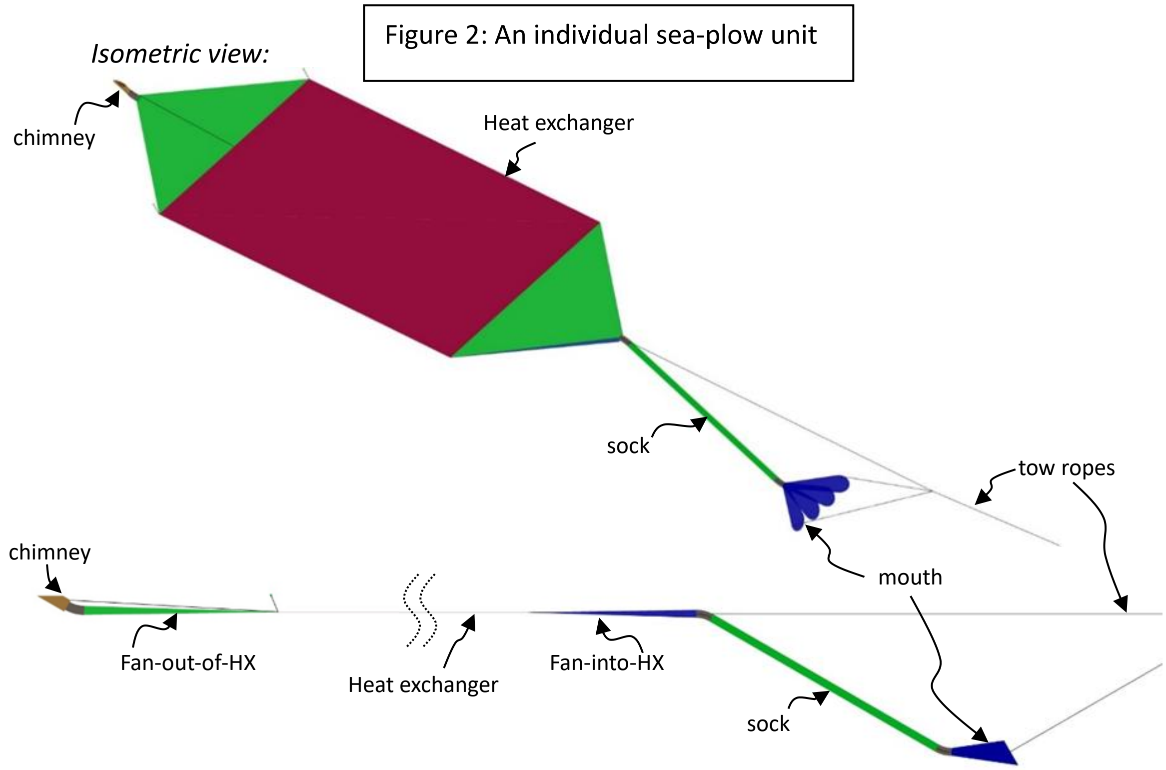
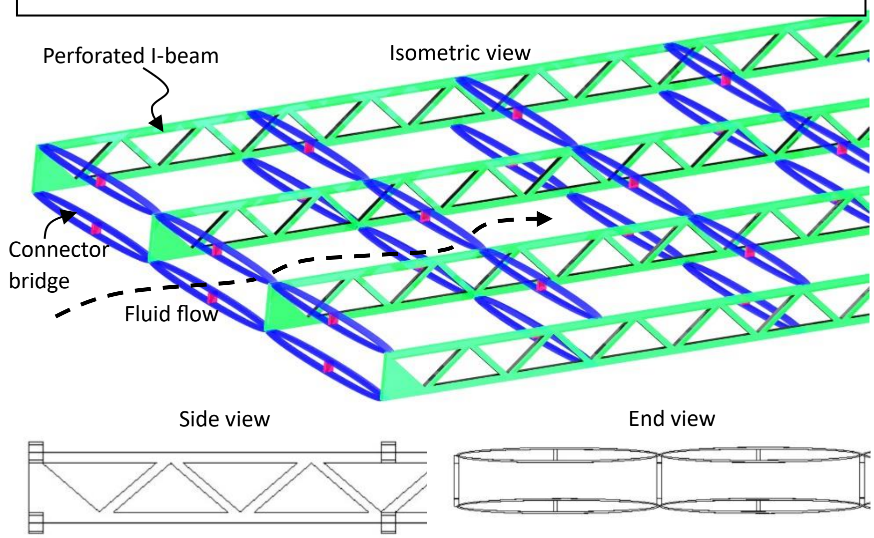


Figure 2 shows an individual sea-plow unit. It is composed of four parts: the mouth, the sock (the tube taking water from 600m to 60m depth), the heat exchanger, and the chimney (the tube taking water from 60m depth to the ocean surface). The parts are designed around the heat exchanger, which is tasked with taking the entrained water from 4 C to 15 C, by holding it about 60m below the ocean surface, in water about 20 C. The water inside the heat exchanger is also designed to flow past it at the same velocity as the water outside it, for convection-calculation purposes. Appendix A details the heat exchangers calculations, which result in its overall dimensions.

Sea-plows surfaces are composed of flexible LDPE glued to a fishnet backing. LDPE is also used in garbage bags. However, garbage bags are 7 mil thick. Sea-plow is using 15 mil thick LDPE. The fishnet is commercially available nylon.

Since the surfaces are flexible, internal pressure would balloon them into shapes which maximize volume, like cylinders and spheres. Restraints are thus needed to force the LDPE into more functional shapes, like the heat exchanger's rectangle. Restraints will also be needed to transmit drag loads and to counter hoop stresses due to internal pressure, so that the surfaces don't tear. The exact nature of these restraints was not designed in this preliminary calculation, but the sizing and costing could be estimated by assuming the restraint was offered by plastic rope. Hence, the restraint calculation is sized in terms of 2-inch diameter polypropylene rope, such as is typically used in marine applications. A strength safety factor of 2 was assumed for the rope. This provided a standard basis for estimating the cost of restraining sea-plows surfaces. As design proceeds, more appropriate restraints will be designed, probably featuring HDPE. However, it is not expected that the overall cost of the restraints will greatly deviate from that calculated here. For the heat exchanger, one possible restraint design is composed of parallel strips of perforated HDPE I-beams, with each I-beam being 0.57m high, as shown in Figure 3. Although different in appearance than an array of rope strands, it is assumed that approximately the same amount of material would be required, and at a similar cost, to restrain the exchanger surfaces into the proper shape.

Figure 3: Perforated I-beam restraint design for heat exchanger



Once the heat exchanger was designed, the other parts could be specified. They are sized to transmit water at the same speed it has in the heat exchanger. The mouth was further designed to develop enough pressure in the sock to push water at the needed speed past the various resistances of the sea plow unit, especially the heat exchanger. Starting from the chimney outflow, at the ocean's surface, where the internal pressure is the same as the surrounding ocean, the rise in internal pressure was calculated, due to the restrictions introduced by each unit. The buildup in internal pressure is due to skin friction and form drag. These drags are a function of water speed, wall area, wall roughness, and the length of each part. At the mouth, the force needed to move water through sea-plow, at the given speed, was calculated to be about 8770kN (or 1750kN/unit). Using Bernoulli's Equation, the mouth area was calculated using $p = F/A = \rho * v^2/2$, where v = 0.7 mph (0.3 m/s). Each unit has a mouth composed of five cones that come together to drive water into the sock (only four cones are shown in Figure 2). The area calculation indicates that each cone should have a mouth diameter of 103m. The five cones fan out from the sock. A maximum 'fan angle' of 40^o was chosen, to keep pressure losses down, which means the mouth fan has a length of 287m, and an overall width of 411m.

The large mouth slows down incoming water and takes it to the pressure needed in the sock to force it the rest of the way through sea-plow. The sock is a tube 32m in diameter. It has an internal truss supporting it, hidden inside the tube. The truss is needed to help maintain the shape of the tube against bending from drag forces as the sock moves horizontally through the ocean. The circular shape of the tube is maintained by its internal pressure. The sock has a 30^o bend at its bottom, and another one at the top. The 30^o angle was chosen to help minimize the drag force associated with turning the entrained water upward toward the surface, since that force is proportional to the angle: $F = A*\rho*v^2*[1-\cos(\text{angle})]$.

The internal truss is composed of HDPE and is designed to be 32m in height, 1080m long, and 0.1m in width. A Neville truss was assumed with elements resembling isosceles triangles (two sides of equal length). The resulting truss was estimated to use 191 m^3 of HDPE in its construction, which figured into its cost estimate.

After leaving the sock, entrained water enters a fan designed to transition it from the tubular sock, 32m wide and tall, to the extremely flattened rectangle of the heat exchanger, 1361m wide and 0.57m tall. This 'fan-into-HX' section is 792m long, which was calculated to maintain a maximum 'fan angle' of 40^o. The shape of the 'fan-into-HX' section is maintained with internal rope restraints on 5m centers, to estimate the cost. It is likely that future design will, as with the heat exchanger, optimize a solution involving perforated HDPE I-beams.

After the entrained water exits the heat exchanger, it again transitions to fit in a 32m diameter pipe, the chimney. This transition occurs in the 'fan-out-of-HX' section, which is identical to the 'fan-into-HX' section in appearance. However, while the 'fan-into-HX' has an internal pressure of 2334 Pa, the 'fan-out-of-HX' has only 19 Pa. The restraint requirement, to maintain its shape, is correspondingly much lower.

The entrained water then enters the chimney, where it is turned upward by a 30^o angle and conveyed from the 60m depth the remaining distance to the chimney exit 5m below the ocean's surface. The chimney also has an associated truss, but it uses only 1% the amount of HDPE as in the sock truss. The chimney doesn't have enough internal pressure for the flexible surface material to hold a cylindrical shape against form drag, as the chimney is pulled horizontally through the ocean. The chimney is thus assumed to form an ellipse, which may be enforced by the internal placement of HDPE structures (not currently included in the calculations).

Appendices

Appendix A: heat exchanger sizing

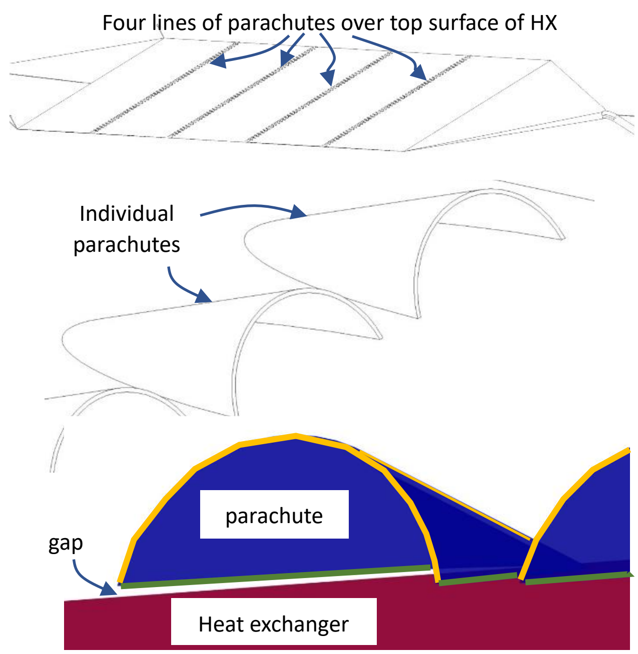
Figure A1 is a snapshot of the heat exchanger design spreadsheet to reference in the following discussion. The best understanding of these calculations comes from the spreadsheet itself. The heat exchanger was sized to take 1173m^3/s of entrained water from 4 C to 15 C, by holding it in ocean water above the thermocline, assumed to be at 20 C. The LMTD (log-mean temperature difference) is 9.5 C. The velocity is 0.3 m/s on the external heat exchanger surfaces.

Figure A1: Heat exchanger sizing calculations

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
1			SeaPlow input parameters:					Depth/Angle of Operation:					Props of Water:					
2	T sea:	20 C		1.0	ppmCO2/year drawdown =			Mouth depth:	600 m				rho	1000	kg/m3			
3	T in:	4 C		4.7E+07	m3/s, upward flow rate req'd			HX depth:	60 m				Cp	4190	J/kg-K			
4	T out:	15 C		40,000	No. of seaplows			Chimney min depth:	5 m					at 4C	at 20C			
5	LMTD:	9.5 C	Qflow:	1173	m3/s/seaplow			Angle off horizontal:	30	deg	0.52 rad		mu (N-s/m2)	0.001537	0.0010195	at Twall avg:		
6	Tavg:	11 C	Qhx:	54045 MW		% of walking speed:			h=Nu*k/L				nu (m2/s)	1.54E-06	1.02E-06	1.17E-06	m2/s	
7	Twall avg:	15 C	v plow:	0.30 m/s	0.7 mph	22%		Nu rough/	h	U			Pr	11.24	7.09	8.32		
8			Outside of HX (a flat plate):	# of tripwires	Re L:	Nu L:		smooth:	(W/m2-K)	(W/m2-K)			k (W/m-K)	0.578	0.602	0.59	W/m-K	
9		L hx:	2000 m	5	1.02E+08	2.06E+05	1.90E+05	1.1	306	210				Props of Plastic wall mat'l:		k polyethylene+graphite:		
10		A HX surface:	27213304	P=A hx/L	Del at Lmax:	19 m	Checks:		U 1st cut	137			k polyethylene:	0.5	W/m-K		1.32	
11		Perim hx:	13607 m	HX chnl dimensions		on Perim	on A chnl	on Qhx:	U final cut	209.7			HX wall thickness:	3.8E-04	m, = 15mil plastic sheeting			
12		A HX channel:	3909 m2	0.57 m	6803 m	13607 m	3909 m2	OK	check:	OK			k/t poly	1312	W/m2-K			
13			HX size:	1.2 mile	4.2 mile								Roughness (ks):	0.005 m				
14		Cf, L:	2.98E-03	1.98E-03											Check if solar heating	Outflow Capacity:		
15		Friction outside HX:	3649 kN												is possible:	4913201	kW/C	
16			Inside of HX (a tube):								Nu multiple:				Solar constant:	0.7	kW/m2	
17		Entrance length:	41 m	Re D:	2.94E+05	h l (Pa/(kg/m3)):	2.29	Nu D:	2.63E+03	1.27E+03	2.1				Delta T:	5	C	
18		Full flow factor:	1.0	f guess	0.030	Δp (Pa):	2294	h (W/m2-K)	1360	h=Nu*k/D						35094293	m2 req'd	
19		D h:	1.15 m	f	0.029	f from NASA:	incropera eqn8.21								Length of surface heater:	73	miles long	
20		e/D:	0.004	f multpl:	2.0	0.014	0.015							Time needed to move seaplow:	7000	miles. Distance N Pacific		
21			Towing Power Calc:	Friction:	Horiz. Force needed		Towing Power Req'd:									Ocean to Southern Ocean		
22			outside	8774 kN	to force fluid upward:		5.4 MW	a large tugboat can tow this							14	months at v plow		
23			inside	9311 kN	47 kN										5	mph: speed needed to make it		
24			Number of Flow Units:	5	check:	Mouth Design (per flow unit):					SeaPlow profile from top:						in 2 months	
25			Total Mouth Input Area:	206918 m2	206992 m2	Diam goal:	100 m	Totl Width:	513 m		Length:	6901 m	4.3 mile	18	mi^2			
26			Mouth Area/unit:	41384 m2		# mouths:	5	Fan angle:	40	deg	Width:	6803 m	4.2 mile					
27	Sock chnl A:	782 m2	Diam sock:	32 m		Diam/mouth:	103 m	length:	287 m					Area of Ocean needed (in % of an Alaska):	109%			

The total length of the heat exchanger is 2000m. The exterior surface resistance to heat transfer is the limiting resistance in this design. Hence, it was decided to induce turbulence on the two exterior surfaces using four lines of 'trip parachutes' per surface, as shown in Figure A2. The parachutes are composed of LDPE held to the surface with rope, and as sea plow is pulled through the ocean the parachutes inflate and break up the boundary layer on the exterior surfaces. The estimated effect of this is to break the exchanger into five sections, each 400m in length.

Figure A2: Trip parachutes on exterior of HX



Assuming a roughness (k_s) of 0.5 cm for the wall material, which is LDPE sandwiched between fishnet, the Nusselt Number was calculated for the exterior by making a rough flat-plate assumption (Ref 14, eqn. 65). This Nusselt Number was 10% higher than a similar calculation assuming a smooth flat-plate (Ref 15, eqn. 7.38), performed as a check.

From this the exterior convection coefficient was derived, which is 306 W/m2-K. The conduction through the 15mil of plastic yields a value of 1312 W/m2-K, and internal convection is likewise also quite high, so the exterior convection resistance was assumed to be the primary resistance in the overall resistance. With it, the derivation of the heat exchanger basic geometry was possible: it's mouth area, perimeter, height and width.

The flat-plate assumption doesn't work for the inside of the heat exchanger, due to the restriction of the growth of the boundary layer by the opposing heat exchanger wall. Instead, a rough tube assumption was made. The hydraulic radius was calculated from the heat exchanger geometry, as well as the relative roughness, and the friction factor and head loss calculated (Ref 16, eqn. 8.37). The Nusselt number was also calculated (Ref 15, eqn. 8.62), leading to the internal convection coefficient, which at 1360 W/m2-K was four times the size of the external convection coefficient. These calculations effectively sized the heat exchanger for the needed heat transfer and gave estimates of the drag on the exchanger surfaces, which internally was calculated as head loss (in Pa).

Appendix B: Internal frictional loss calculations

Figure B1 is a snapshot of the spreadsheet making frictional loss calculations, both internal and external. The internal calculation is of head loss since the interior of sea-plow resembles a variously shaped pipe. This calculation begins at the mouth and ends at the chimney.

Figure B1: Frictional loss calculations

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
28																	
29	**Internal Δp Calculation:**			14.1091121		Total internal Δp (Pa):	2382	0.35	psi	**External Friction Calc:**		per flow unit		Total External Drag:	1755 kN	per flow unit	
30		**Mouth Fan:**			**Sock bend 1:**			**Fan into HX:**			**Mouth Fan skin drag:**						
31		v avg fan:	0.15 m/s		Diam:	32 m		Width/HX unit:	1361 m		Re L	5.60E+07		**HX F** skin F:	730 kN	per flow unit	
32		D avg fan:	58 m		e/D:	0.0002		Fan Length:	792 m		Skin Area:	263247 m2					
33		e/D:	0.0001		ReD:	8.07E+06		Avg fan angle:	0.41 rad		Cf, L:	3.15E-03	2.09E-03	**Fan out of HX skin drag:**			
34		ReD:	5.70E+06		f guess	0.013		bend radius (Rb):	1952 m		Skin F:	37 kN		Re L:	2.03E+08		
35		f guess	0.012		f	0.013		Rb/D:	61.9					Skin Area:	1117149 m2		
36		f	0.012		f multpl:	1.7		Bend angle, deg:	23		**Sock form drag (assumes circle):**			Cf, L:	2.67E-03	1.79E-03	
37		f multpl:	1.4		bend L:	60 m	input:	61.9	23.2		Frontal Area:	17036 m2		Skin F:	134 kN		
38		bend radius (Rb):	411 m		bend radius (Rb):	115 m	Rb/D Deg:	20	30		ReD:	6.16E+06					
39		Rb/D:	7.0		Rb/D:	3.6	20	0.045	0.073		Cd:	0.67		**Chimney Ellipse:**			
40	input:	7.0	20.0		Bend angle, deg:	30	20	1	1		Multiple:	1.2		Area:	782 m2		
41	Rb/D Deg:	20	20	input:	3.6	30.0		0.045	0.073		drag:	616 kN		D:	45 m		
42	6	0.045	0.045	Rb/D Deg:	30	30		kb:	0.054		**Sock skin drag:**			Perim:	110.8 m		
43	6	0.045	0.045	3	0.098	0.098		hl, bend (Pa/(kg/m3)):	2.2E-02		Length:	1427 m					
44		0.045	0.045	4	0.08	0.08		Fan into HX Δp (Pa):	21.5		Re L:	2.79E+08		**Chimney form drag (assumes ellipse):**			
45		kb:	0.045		0.08662067	0.086620673		HX Δp (Pa):	2294		Skin Area:	59468 m2		Area=b*D:	2454 m2		
46		hl, bend (Pa/(kg/m3)):	1.0E-03		kb:	0.087		Fan out of HX Δp (Pa):	21.5		Cf, L:	2.43E-03	1.64E-03	ReD:	8.71E+06		
47		Mouth Fan Δp (Pa):	1.04		hl, bend (Pa/(kg/m3)):	6.3E-03		Chimney Bend Δp (Pa):	6.3		Skin F:	7 kN		Cd:	0.17		
48		Entered into total Δp:	1.04		Sock Bend 1 Δp (Pa):	6.3		**Chimney:**						Multiple:	1.2		
49		Check:	OK		Sock Bend 2 Δp (Pa):	6.3		Chimney L:	80 m		**Fan into HX form drag**			drag:	23 kN		
50								hl (Pa/(kg/m3)):	1.5E-03		Fan "b*D"	558574 m2					
51				**Sock:**	Sock L:	1020 m		Chimney Δp (Pa):	1.5		"D" (i.e. L):	411 m		**Chimney skin drag:**			
52					Entrance L:	1966 m					ReD:	8.01E+07		Re L:	4.12E+07		
53					entrance multiple:	1.25					Cd:	0.006		Skin Area:	9419 m2		
54					hl (Pa/(kg/m3)):	2.4E-02					Multiple:	1.4		Cf, L:	3.57E-03	2.35E-03	
55					Sock Δp (Pa):	23.8					drag:	207 kN		Skin F:	2 kN		
56																	

The mouth has five 103m diameter tubes channeling entrained water into one 32m diameter tube, the sock. Since the head loss is the same in each of the tubes, only one tube's head loss was calculated. For this tube, an 'average bend' was assumed, to include the small effect of the mouth fan bending entrained water into the sock entrance. The average velocity in this tube is half the speed of sea-plow since the water at the mouth entrance is essentially stagnant. The relative roughness (e/D) is used: it's estimated the friction factor is 40% higher than for a smooth tube. Losses due to the bend were calculated (Ref 11, eqn 1). Also, an entrance multiplier of 1.25 is used to account for entrance effects.

The sock is a tube 32m in diameter, 1020m long, with two 30^o bends in it, each with a bend length of 60m. Head losses for the bends (Ref 11, eqn 1), at 6 Pa each, were calculated separately from the straight section, which was 24 Pa. Roughness and entrance effects are applied to all losses. Losses are about 70% higher due to roughness compared to a smooth pipe.

The 'Fan-into-HX' and 'Fan-out-of-HX' have identical head losses. Water in these fans is bent by an average angle, which is half the maximum angle of 40^o chosen for the fan. Effects of bending, roughness and entrance are included.

The internal head loss in the heat exchanger, 2294 Pa, is calculated in Appendix A, in which the heat transfer requirement determines its overall size. Roughness doubles its friction factor over a smooth pipe. The losses are based on pipe flow using a hydraulic diameter, D_h = 4 Area/Perimeter (Ref 16, eqn. 8.37). The heat exchanger cross-section in this case is highly rectangular. A NASA paper (Ref 12) which included formulae validated for calculating friction in rectangular channels was used to check the standard calculation. From these results, as discussed also in Appendix A, the internal Reynolds Number and Nusselt Number were calculated yielding the internal head loss and convection coefficient.

The chimney has a single 30° bend in it, at its entrance. The bend length is 60m, which given the vertical distance to the ocean surface meant the straight section was 80m long. Head losses in the bend were assumed equal to that previously calculated for the sock bend. For the straight section a friction factor equal to that used in the sock straight section was used on its much shorter length.

Appendix C: External friction calculations

Internal sea-plow friction is due to skin drag from flow parallel to the skin. External friction, in addition to skin drag, also includes form drag, from flow normal to external surfaces and forced to deviate around them. Form drag only applies to the sock and chimney (columns extending vertically against a horizontal flow), and to the 'fan-into-HX', which leads the HX assembly of parts into the oncoming water. Refer to Figure B1 for external friction calculations.

The mouth fan is subject to external skin drag only, proportional to its surface area, velocity, roughness (k_s) and length.

The sock was assumed to be a vertical cylinder for form drag calculations, and then skin drag calculations were made on the skin area not included in that assumption. For a smooth cylinder normal to the flow the drag coefficient (C_d) is a function of Reynolds Number, which indicated full turbulence (Ref 13, fig. 9.21). This C_d was increased by 20% to account for roughness. The drag coefficient is then multiplied by the frontal area and $\frac{1}{2}\rho v^2$ to yield the external force. This turned out to be about half the total external drag on each sea-plow unit. The additional skin drag due to the additional sock area unaccounted in the form drag calculation was 1% of the form drag.

The 'fan-into-HX' occurs at the leading edge of the heat exchanger assembly, which is otherwise dominated by skin drag. The 'fan-into-HX' part, however, was modelled using the form drag calculation on a flat plate (Ref 13, fig. 9.22). The drag coefficient (C_d) from this calculation was increased by 40% due to roughness.

The heat exchanger's external friction coefficients were worked out previously in Appendix A. Given the area of the heat exchanger, and accounting for roughness, about one-third of the total external resistance is due to the heat exchanger.

The 'fan-out-of-HX' trails the heat exchanger, and its area adds its skin drag to the total. This is calculated to be about 33% lower than that of the 'fan-into-HX.

The chimney has the form of an ellipse, due to its low internal pressure. It is held rigid against the drag by an internal truss but may require further internal HDPE reinforcement to maintain the ellipse shape. The drag coefficient (C_d) of the chimney ellipse was estimated (Ref 13, fig. 9.22). It was then increased by 20% to account for roughness. An additional 5% of skin drag was added to this, since the chimney is not vertical, but slanted 30^o, and has extra skin area for frictional losses.

Appendix D: Sizing the trusses, tow ropes, and axial loads.

Refer to Figure D1 in the discussion to follow. The trusses are designed to take horizontal bending loads on the sock and the chimney, which extend over significant vertical spans. For ease of sizing the trusses Neville trusses were assumed. Each truss section is an isosceles triangle, i.e. having two sides of equal length, with an internal angle, α. The upper chord, upon which the load is applied, is composed of the

Figure D1: Truss Design

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
56																		
57	**Sock Truss Design:**				Y.S. HDPE/S.F.:	11	MPa, S.F.:	2			**Chimney Truss Design:**							
58	Length:	1080 m	Height:	32 m	truss internal angle (deg):						Length:	110 m	Height:	1.1 m	truss internal angle (deg):			
59	= F/L (in kN/m):	0.3	Width:	0.10 m	59.6	1.04 rad					w = F/L (in kN/m):	0.11	Width:	0.03 m	60	1.05 rad		
60	f	Fs (kN)	Fs height	Fa, b (kN)	Fa,b height	Vol Fs:	Vol Fa+Fb:			f	Fs (kN)	Fs height	Fa,b (kN)	Fa,b height	Vol Fs:	Vol Fa+Fb:		
61	0	179 kN	0.16 m	89 kN	0.08 m	7.08	3.5			0	7 kN	0.03 m	3 kN	0.03 m	0.04	0.04		
62	0.25	134 kN	0.12 m	649 kN	0.59 m	5.31	25.5			0.25	5 kN	0.03 m	68 kN	0.21 m	0.04	0.27		
63	0.5	90 kN	0.08 m	1043 kN	0.95 m	3.54	41.0			0.5	3 kN	0.03 m	115 kN	0.35 m	0.04	0.46		
64	0.75	45 kN	0.04 m	1270 kN	1.15 m	1.77	49.9	No of Fs:	59.8	0.75	2 kN	0.03 m	142 kN	0.43 m	0.04	0.57	No of Fs:	173
65	1	0 kN	0.03 m	1331 kN	1.21 m	1.30	52.3	Total HDPE in Truss:		1	0 kN	0.03 m	151 kN	0.46 m	0.04	0.60	Total HDPE in Truss:	
66			f max:	Fb max		19	172	191	m3			f max:	Fb max		0.2	1.9	2.1	m3
67			0.967	1333 kN								0.988	151 kN					
68	**Effect of Internal Pressure on Sock and Chimney Shapes:**								**Tow Rope Tensions/flow unit**					**Axial Forces/flow unit**	internal	external		
69		Int'l Pressure	w (kN/m) from:		Conclusion: Internal pressure will keep Sock circular,				F tow2:	3241 kN	F tow3 ang	3.5	deg	Chimney	6 kN	NA		
70		Pa	Int'l Press	Ext Drag	but chimney will form an ellipse and probably				F tow1:	445 kN	F tow3:	3633 kN		Fan out HX	17 kN	134 kN		
71	Sock	2302	73	0.3	require some internal structure to maintain				Sum (horiz):	3627 kN	F tow4 ang	1.9	deg	HX	1793 kN	730 kN	Check	% of F tow2
72	Chimney	1.5	0.03	0.11	that shape (not included in current design).				Check:	3627 kN	F tow4:	24 kN		Fan into HX	17 kN	0 kN	2898 kN	89%
73														Sock	28 kN	548 kN		% of F tow1
74														Mouth	1 kN	37 kN	704 kN	158%
75																		99%

'a' struts, while the lower chord is composed of the 'b' struts. The web is composed of struts that slant between upper and lower chords, which are the 's' struts. Since the normal load is symmetrical, only half of the truss needs to be considered, and the load is calculated according to the fractional length, f, where f =0 at one of the truss edges, and f=1 at L/2, where L is the length of the truss. The force on the webs slanted struts is $F_s = (wL/2)*(1-f)/\cos(\alpha/2)$, where α is the internal angle, L is the truss length, and w is the drag force per length normal to the truss. Alternate slanted struts are in tension or compression (tension if, at the location in question, the slanted strut is incoming to the top chord, and compression if incoming to the bottom chord). The maximum slanted strut force occurs at f =0, i.e., at the pinned edges of the truss. By summing moments about the nodes along the bottom chord 'b', the loading of the top chord 'a' can be calculated: $F_a = (wL^2/(8h))*(2f - f^2) + \{1 \text{ or } 0\}*F_s*\sin(\alpha/2)$. The {1 or 0} depends on the location along the truss ("1" applies if, at that location, a slanted strut is incoming to the top chord, the "0" if it is incoming to the bottom chord). This compressive force is maximum at the center of the truss, where f =1. The bottom chord, 'b', is in tension. For the bottom chord, $F_b = F_a + \{-1 \text{ or } 1\}*F_s*\sin(\alpha/2)$. For simplicity, both top and bottom chords were sized to include the "$F_s*\sin(\alpha/2)$" term, i.e. $F_{a\,max} = F_{b\,max} = (wL^2/(8h))*(2f - f^2) + (wL/2)*(1-f)*\tan(\alpha/2)$. To estimate the cross-sectional area (A) for each strut, A = F/(Y.S./S.F.), where F = F_s, F_a, or F_b, Y.S. = yield strength of HDPE (22MPa), and S.F. = safety factor, chosen as 2. The length of each strut is known from geometry, so in this way the truss material requirement was obtained and the cost estimated.

Each unit of sea-plow has four tow ropes external to it, and together they tow the unit horizontally through the ocean (see Figure D2). 'Tow1' pulls the mouth through the ocean. It has a 30^o angle to horizontal. Thus the mouth should be canted in such a way as to encourage the mouth to dig down into the ocean as it is pulled through it to maintain its vertical position. 'Tow2' pulls the heat-exchanger assembly horizontally through the ocean. Tow2 connects to Tow1 about 1.3 miles in front of the unit. 'Tow3' pulls these two through the ocean and is connected to the towing vehicle. Tow3 is angled 4.1^o with the horizontal. Tensions in each of the tow ropes is calculated in Figure D1.

Tow2 was sized to maintain zero horizontal moment about the unit at the vertical location of the mouth (600m). Since Tow2 is applied at the depth of the heat exchanger (60m), its moment arm is 540m:

$\Sigma M_{mouth} = 0 = F_{chimney}*570m + F_{hx}*540m - F_{tow2}*540m + F_{sock}*(540m/2)$

These are all centered external drag forces, except for F_{hx}, which should include the internal drag of all sections up to the entrance to the 'fan-into-HX' section: F_{hx} = external drag on HX and two fans + $\Delta P*A_{sock}$, where ΔP includes all sections from the chimney to the 'fan-into-HX', and Asock = $\pi*D_{sock}^2/4$.

Tow1 is the difference between the total horizontal force per unit and Tow2, and Tow3 was sized to the combined forces of Tow1 and Tow2.

'Tow4' connects the chimney to the back of the heat exchanger, to maintain the chimney's vertical position against the horizontal drag force. The tension on Tow4 is equal to the chimney drag, moderated slightly by its small angle to the horizontal, which is 2.2°.

Figure D2: Tow ropes: 1,2 and 3

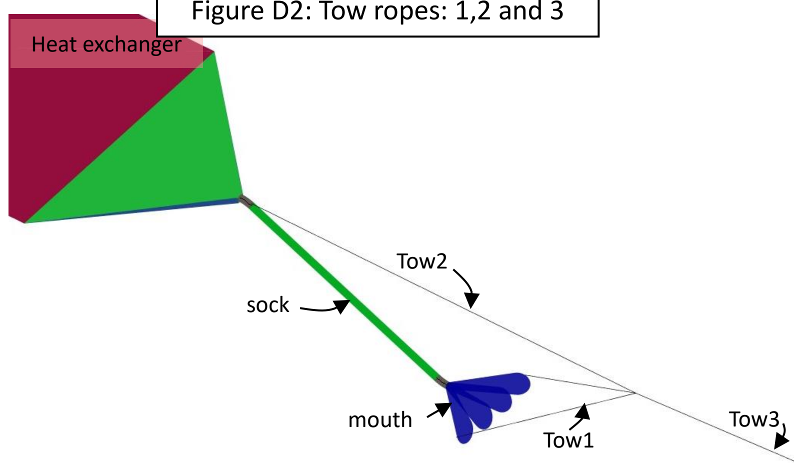


Figure D1 also shows the derivation of the incremental axial loads imposed by each section of sea-plow. These incremental loads are summed up in estimating the axial rope requirement needed in each section to keep the surface from tearing. Again, this is for an estimate of the cost of this restraint, and not the recommended design solution for the restraint. The internal axial force per section is the pressure times the cross-sectional area of that section. For most sections, this cross-sectional area is the area of the sock: 782 m^2. Since the force is incremental, the pressure is really ΔP, the head loss. The external axial load is the external friction, previously calculated in Appendix A. One exception is for the 'fan-into-HX', where this external friction is primarily a form drag that acts to reduce the tensioning of the ropes that counter axial loads. A zero axial load was chosen for this part. For the sock, external friction (mostly from form drag) has components normal to, and aligned with, the sock itself. Therefore, the sock angle must be factored in to calculate the axial load on the sock. The normal load is countered with the internal truss.

Appendix E: Estimating sea-plows restraint and surface requirement.

In all cases, restraints were assumed to be supplied by 2” diameter polypropylene rope, such as is commonly found in marine applications. This was for ease of calculating this cost, and not as a recommendation for a final design solution. The rope requirement was then doubled, to reflect a factor of safety of 2. As indicated in Figure E1, there were four categories of ropes: Interior, towing, circumferential, and axial. Interior ropes are needed to maintain the general shape of a section of sea-plow, such as the heat exchanger. Without these restraints, internal pressure would blow the flexible skin material into a cylinder, a sphere, or similar shape that maximizes internal volume per surface area. The three sections of sea-plow that needed these restraints were the heat exchanger, and the two fans surrounding it. The internal restraint requirement is directly related to the average internal pressure in each section, which is previously calculated. Once a rope spacing is selected, the requirement is calculated directly from the pressure and the area between ropes.

Figure E1: Rope, surface, and buoy calculation

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T
76	**Rope requirement per seaplow**			Safety factor:	2	TABLE: 2"	rope ($/mile)	Break streng	Spec Grav											
77			choice	$/mile	Strength:	nylon	20064	409 kN	1.11	link	link2									
78			polypropylene	8835	104 kN	polypropylene	8835	208 kN	0.88	link	link									
79																				
80			Interior ropes:				Towing ropes:					Circumferential ropes:					Axial ropes:			
81			Fan into HX	HX	Fan out HX	Tow 1	Tow 2	Tow 3	Tow 4	Buoy	Mouth	Sock	Fan into HX	Fan out HX	Chimney	Fan out HX	HX	Fan into HX	Sock	Mouth
82		Δp (Pa):	2334	1176	19	NA	NA	NA	NA	NA	1191	2382	2334	19	NA	NA	NA	NA	NA	NA
83		Spacing:	5 m	5 m	10 m	NA	NA	NA	NA	NA	5 m	10 m	5 m	10 m	37 m	100 m	100 m	100 m	33 m	42 m
84		Ropes/flow unit:	16750	108844	4187	2	1	1	1	2	287	114	40	20	3	14	27	14	3	22
85		Ropes/sea plow:	83750	544220	20935	10	5	5	5	10	1435	570	198	99	15	71	136	71	15	109
86		Force/rope:	58 kN	29 kN	2 kN	223 kN	3241 kN	3633 kN	24 kN	18 kN	167 kN	376 kN	184 kN	3 kN	2 kN	5 kN	53 kN	190 kN	96 kN	24 kN
87		No. of 2" ropes needed:	0.6	0.3	0.0	2.1	31.1	34.9	0.2	0.2	1.6	3.6	1.8	0.03	0.02	0.05	0.51	1.83	0.92	0.23
88		diam of combined rope:	1.5 in	1.1 in	0.3 in	2.9 in	11.2 in	11.8 in	1.0 in	0.8 in	2.5 in	3.8 in	2.7 in	0.3 in	0.3 in	0.5 in	1.4 in	2.7 in	1.9 in	1.0 in
89		Rope length:	0.0023	0.0004	0.0023	0.7 mile	1.3 mile	0.6 mile	0.5 mile	0.04 mile	0.11 mile	0.06 mile	0.06 mile	0.06 mile	0.07 mile	0.49 mile	1.24 mile	0.49 mile	0.67 mile	0.18 mile
90		2" rope/seaplow:	107 mile	55 mile	1 mile	14 mile	209 mile	106 mile	0.6 mile	0.06 mile	263 mile	127 mile	22 mile	0.2 mile	0.02 mile	1.8 mile	85.8 mile	63.4 mile	9.3 mile	4.4 mile
91																				
92	**Interface Surface requirement (per flow unit):**											**Buoy sizing:**								
93			*LDPE plastic:*				*LDPE for HX:*					Chimney weight	1.0 kN			Buoy Volume Safety Factor	2			
94		Mouth	263247 m2	Fan out HX	1117149 m2	HX	5442661 m2					Chimney weight/buoy	0.5 kN			Volume/buoy:	3.7	m3		
95		Sock	112990 m2	Chimney	12190 m2							Force/buoy	18 kN	2.8%		Height of 1m Diam buoy	4.7 m			
96		Fan into HX	##########	**Total:**	2622724 m2							Displaced water/buoy	1.8	m3		Diam of 4m high buoy	1.1 m			
97																				

Because it has the highest pressure, the ‘fan-into-HX’ section has the longest rope requirement: 107 miles (of 2” diameter rope). The heat exchanger requires 55 miles, and the ‘fan-out-of-HX’ requires 1 mile. As mentioned previously, the ideal solution for this kind of internal restraint will likely not involve rope at all, but it is assumed here that its eventual cost is reflected in these rope requirement estimates.

Towing ropes connect sea-plow to the tugboat, or other towing device. The tension on these tow ropes was calculated previously; here that tension is used to determine the rope requirement in terms of 2” diameter rope. Tow 2 has the largest requirement as it is the longest tow rope and is pulling the majority of sea-plow.

The primary sea-plow surface is composed of flexible LDPE glued to a fishnet backing. This is assumed to be sufficient to handle local stresses related to its function as a physical barrier. Larger stresses are handled using ropes attached to the outside of the primary surface, to the fishnet backing. The amount of rope required for this structural function is, for cost-estimation purposes, expressed in an amount of 2” diameter polypropylene rope for each section. Circumferential ropes surround the section in a direction normal to the movement of entrained water, and axial ropes surround it in a direction parallel to that movement. Circumferential ropes counter hoop stresses and so are primarily a function of internal pressure and section size. Axial ropes transmit drag forces along the direction of movement, so the axial forces previously determined are used in estimating this rope requirement. For example, for the heat exchanger, the axial force associated with the entire ‘fan-out-of-HX’ section and half of that of the heat exchanger itself is used in sizing the axial rope requirement. For the ‘fan-into-HX’, the axial forces for the entire ‘fan-out-of-HX’ and heat exchanger sections are used along with half of the axial force of the ‘fan-into-HX’ itself.

Sea-plows estimated surface requirement (LDPE bonded to fishnet) comes from the overall dimensions of its parts. The fishnet is also expected to cover the water entry and exit areas of sea-plow, to prevent fish from getting caught up in its interior. As shown in Figure E1, the heat exchangers surface area is listed separately. This is in case a special LDPE material should be needed there, to meet heat exchange requirements (for example, there are versions of polyethylene that are infused with graphite nanoparticles to raise their thermal conductivity). A special LDPE material was found not to be needed with the current design.

Two buoys per heat exchanger are baselined, mostly to help maintain the proper depth for sea-plows parts: 600m for the mouth, and 60m for the heat exchanger. The buoys are positioned, one at each side of the heat exchanger, near the back end. Primarily, they are sized to resist the downward force of the water entrained by sea-plow that has been accelerated upward toward the surface, which is estimated to be 18kN per buoy. Each buoy is 1m in diameter and 4.5m high, with a buoyancy safety factor of 2 (see Figure E1). Figure E2 shows a potential design for these buoys, designed with a surrounding bumper to protect smaller vessels from collisions, and with a water-brake positioned 50 ft or more below the buoy. The brake is intended to pull the buoy through the ocean in large wave conditions, thus ensuring a minimal effect of wave action on the sea-plow parts below.

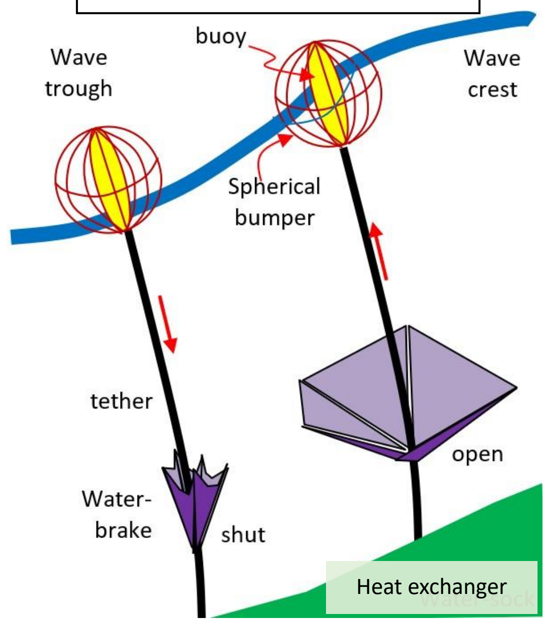


Figure E2: Buoy design

Appendix F: Economic calculations

Figure F1 shows the economic calculations for sea-plow. All costs were annualized to put them on an equal footing. A useful life of 22 years was assumed: 2 years to build sea-plow and 20 years of operation. With an interest rate of 4%, the capital recovery (amortization) factor $A_{t,r}$, was 14.5. Capital costs to which this applied were the cost of purchasing the tugboat, and sea-plow's construction costs.

Figure F1: Economic Calculations

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
98	***Calculate Cost per year per plow:***								**Example cost of trusses:**						**Material cost of Truss HDPE:**			
99	**Economic Parameters:**		Useful life:	22	=2yrs construction+ 20yrs in the water					$ 400	for 60 ft long truss, over a 10 ft expanse				HDPE:	density	970	kg/m3
100			Interest rate:	4%					sock is:	59	60ft trusses in length&		10	in width		cost	1.00	$/kg
101			A t,r:	14.5					sock is:	611	trusses overall (each 60 ft long)					Truss: Sock	Chimney	ratio:
102									cost ($):	370,674					Weight (kg)	185405	2076	1.011
103		**Cost per SeaPlow:**		Cost/unit	Unit										Mat'l cost	$ 185,405	$ 2,076	
104		*Material costs:*	unit	($/unit)	requirement	Cost (mill$/yr)			***Example costs of fishnet and tarp:***						total cost ratio:	51%		
105		LDPE (non-HX)	m2	0.9	1.3E+07 m2	0.8				Area ft2	m2	Cost	$/m2	link		So, est cost is likely conservative		
106		LDPE (HX)	m2	0.9	2.7E+07 m2	1.8				150	13.9	$ 91.50	6.6	westcoastnetting				
107		Fishnet	m2	4.7	4.1E+07 m2	13.2				225	20.9	$ 117.70	5.6					
108		2"D rope	mile	8,835	1069 mile	0.65				400	37.2	$ 184.94	5.0	thenauticalplace				
109		Buoys	buoy	15,393	10	0.01				720	66.9	$ 111.00	1.7	floridanets				
110		Trusses	both trusses	370,674	5	0.13				Example blue tarp costs/area ($/ft2):			0.07	0.09	blue tarps	0.09	tarpsplus	
111					Total Material Cost	17			***Example cost of buoy:***									
112			Assume construction/deployment is the same			17				$ 322.6	for 30"D Taylor Made Products, Sur-Moor T3C Mooring Buoy, Seamless, Foam Filled							
113					Towing cost	21				0.23	m3, for 30"D buoy							propulsion
114					**Cost per year per plow:**	**54**	**mill $/yr**		***Example cost of tugboat:***			mill $/year	2 MW diesel generator: 141.9gal/hr				price/gal	efficiency
115											Cost of diesel:	16.9	2	MW	1243044	gal/yr	$ 3.50	70%
116					**For all sea-plows:**	2	trillion $/ year				Tugboat purchase:	0.17	$ 2,390,000	purchase cost				
117						218	GW of towing power req'd				Tugboat O&M:	4.0	compare to: DP-2 vessel:					
118											Total:	**21.1**	**9.8**		Diesel/total:	80%		
119					Sequestered C:	2.13E+09	metric tons/yr											
120						$ 1,019	per ton C sequestered											
121						$ 278	$/ton CO2 sequestered											

Sea-plow requires 5.4 MW of towing power. This seems a reasonable requirement as tugboats in port are usually less than 2.5 MW, but deep ocean tugboats of up to 20 MW exist. Tugboats are available from $36000 to $2.4 million. Assuming the latter cost in this application, the annual cost for a 22 year life at 4% interest is $0.17 million. Assume an annual Operations & Maintenance cost of $4 million. It is expected that a 2 MW diesel generator uses 1,243,044 gallons of diesel a year, which at the current wholesale cost of $3.50/gallon, and rated up to 5.4 MW for this application, and assuming a propulsion efficiency of 70%, comes to $16.9 million annually. Thus, a total towing cost of $21.1 million per year was used in the economic evaluations. For comparison, the cost of using a DP-2 vessel for placement of an ocean impeller device is estimated at $10 million per year, but it is likely that that application uses much less fuel than this would (Ref 17, Table 6-13).

A surface material requirement of 13 km^2 of LDPE is needed for sea-plow (all sections except the heat exchanger). At a cost of $0.90/m^2, and dividing by $A_{t,r}$, this cost is $ 0.8 million per year. The per unit cost of LDPE is for 'hurricane rated' tarp material, 15mil thick.

The heat exchanger is currently designed to be composed of ordinary LDPE. It is accounted for separately in case a different material is needed to enhance its thermal conductivity. The heat exchanger has a surface area of 27 km^2.

Glued to the LDPE is nylon fishnet. This is for structural purposes. The fishnet is also used alone to cover the entrance and exit to sea-plow to prevent fish from entering. The material requirement for fishnet is 41 km^2, and a unit cost of $4.70/m^2 was assumed.

This initial design for sea-plow is using 2" diameter polypropylene rope as the generic 'restraint' material used to enforce its surface outlines and reinforce the surface against internal pressure and drag. The total amount of this material is assessed to be 1069 miles. A cost of $8,835/mile is assumed in this analysis.

Each buoy on sea-plow is assumed to cost \$15,390, based on smaller buoys on sale. There are two buoys per heat exchanger, thus 10 per sea-plow.

Trusses for large buildings cost about \$400 for a 60ft length, and 10ft width. About 611 such trusses would be needed on each sea-plow unit. Doubling the material cost of the truss, to account for labor, a total cost of \$370,000 was derived.

The two trusses are composed of HDPE. Assuming a cost of \$1.00/kg, the material cost comes to \$185,400. Doubling this for labor, a similar value is achieved to that above. Hence, the \$370,000 cost for the two trusses was used in the economic analysis.

Added everything up, the material cost for sea-plow comes to \$17 million/year. It was assumed that the annualized cost of the labor needed to assemble these materials into a deployable sea-plow structure, and to deploy it in the ocean, is similar. The cost of towing comes to \$21 million/year, 77% of which was the cost of the diesel fuel. Each sea-plow comes to \$54 million/year to operate.

The original goal was to use sea-plow to lower the current atmosphere's CO2 content by 1ppm/year. The atmosphere weighs 5.15x10^{21}grams. Its molecular weight is 28.97, so this is (5.15/28.97)x10^{21} moles of gas molecules. Of this, 0.0004 (i.e. 400ppm) contain Carbon as CO2, giving a total of 0.0004*(5.15/28.97)x10^{21} moles of Carbon in the atmosphere. To remove 1ppm of this each year is a sequestration rate of (1/400 per year)*[0.0004*(5.15/28.97)x10^{21} moles of Carbon], which is 1.78x10^{14} mol C/yr.

It is assumed that the upwelled water will fertilize the ocean such as to eventually sequester 0.12 moles C/m3 of upwelled water (Ref 5), which for 40,000 sea-plows, requires an upwelling rate of 1.78x10^{14} mol C/yr /0.12 mol C/m3 water /40,000 sea-plows = 1175 m3 water/s per sea-plow. The size and speed of sea-plow is designed around this upwelling requirement.

Each year, a single sea-plow would sequester 1.78x10^{14} mol C/yr /40,000 sea-plows = 4.44x10^{9} mol C/yr/sea-plow. Sequestration technologies are compared in units of tons of carbon sequestered per year, so per sea-plow this is 4.44x10^{9} mol C/yr/sea-plow *(12 gm/mol C)*(10^{-6} metric ton/gm) = 5.33x10^{4} metric tons/year/sea-plow. With each sea-plow costing \$54 million/year to operate, the cost is \$1019/metric ton of C sequestered. This is \$1019*(12/44) = \$278/metric ton of CO2 sequestered, since the molecular weight of CO2 is 44.

\$278/ton of CO2 sequestered remains above the desired value of \$100/ton. There's some conservatism in the cost assumptions that make it possible to bring this estimate down with more experience and with economies of scale.

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