A cost-effective steam-driven RO plant for brackish groundwater

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Abstract: Desalination is a costly means of providing freshwater. Most desalination plants use either reverse osmosis (RO) or thermal distillation. Both processes have drawbacks: RO is efficient but uses expensive electrical energy; thermal distillation is inefficient but uses less expensive thermal energy. This work aims to provide an efficient RO plant that uses thermal energy. A steam-Rankine cycle has been designed to drive mechanically a batch-RO system that achieves high recovery, without the high energy penalty typically incurred in a continuous-RO system. The steam may be generated by solar panels, biomass boilers, or as an industrial by product. A novel mechanical arrangement has been designed for low cost, and a steam-jacketed arrangement has been designed for isothermal expansion and improved thermodynamic efficiency. Based on detailed heat transfer and cost calculations, a gain output ratio of 69–162 is predicted, enabling water to be treated at a cost of 71 Indian Rupees/m^3 at small-scale. Costs will reduce with scale-up. Plants may be designed for a wide range of outputs, from 5 m^3/day, up to commercial versions producing, 300 m^3/day of clean water from brackish groundwater.

Keywords: Reverse osmosis; Rankine cycle; Recovery ratio; Gain output ratio; Solar; Biomass

Symbol	**Unit**	**description**
A_{mem}	m^2	Area of RO membrane
c	kmol/ m^3	Molar concentration
Cc	INR	Capital cost (Indian Rupees)
C_{pf}	kJ/ kgK	Specific heat of water
C_{pg}	kJ/ kgK	Specific heat of steam
d_p	m	Diameter of power piston
d_w	m	Diameter of water piston
E	kJ	Energy
F_{g12}		Grey body factor
h	W/ m^2 K	Convective heat transfer coefficient
i		Interest rate
INR		Indian rupee
L_p	m	Length of power stroke
L_w	m	Length of water stroke
MA		Mechanical Advantage
n	years	Life span of machine

P_{net}	bar	Net driving pressure
P_{osmf}	bar	Osmotic pressure of feed water
Q	litres/ s	Flow rate of treated water
q_{in}	kJ	Energy input
q_{out}	kJ	Energy output
R	INR	Repayment installment
R_p	m	Length of power side crank
R_w	m	Length of water side crank
s	m/ sPa	Permeability
T_c	^oC	Condenser temperature
T_h	^oC	Input steam temperature
V	m^3	Volume
X_p	m	Movement of power piston from cylinder end
X_w	m	Movement of water piston from cylinder end
σ	$W/m^2 K^4$	Stefan- Boltzmann constant
ε_1		Emissivity of wall
ε_2		Emissivity of steam
α_p	degree	Crank angle of power side crank from horizontal
α_w	degree	Crank angle of water side crank from horizontal
η		Efficiency
''		Inch
GOR		Gain output ratio
RO		Reverse osmosis
ppm		Part per million

1. Introduction

Availability of usable water has emerged as one of the most crucial problems throughout the globe in recent times. Desalination using reverse osmosis or distillation is a potential solution, but economic and environmental constraints limit the use of such technologies [1]. In developing and underdeveloped countries, the situation is especially severe, as there is usually a pressing need for clean water. But high operating costs and energy demands typically rule out desalination [2]. A sustainable and non-polluting solution to energy and water shortages could only be provided by renewable energy systems [3].

In India, availability of water is reducing steadily as population grows. It is estimated that by 2020 India will become a water-stressed nation. Groundwater is the major source of water in the country, with 85% of the population dependent on it. While the urban water supply predominately uses surface and ground water, nearly 70% of drinking water requirements in rural India are met by ground water. The quality of ground water is variable and fails to meet the drinking water requirements in many areas. Frequently it is brackish or contaminated with excess fluoride, arsenic, iron, or microorganisms. In order to augment the available and accessible water, it is necessary to implement suitable treatment technologies to render such contaminated water potable [3].

Prompted by the growing crisis, a technology mission on Winning, Augmentation and Renovation (WAR) for water has been initiated under the directives of the Supreme Court of India to the Union Ministry of Science and Technology. The mission will undertake research-led solutions on a 'war footing' through a national and coordinated approach. WAR for Water has been developed on the principle that timely, urgent, cost effective, socially viable and sustainable techno-management solutions are required to the problems of water scarcity [3].

Desalination is an energy intensive process. In desalination based on membrane Reverse Osmosis (RO), energy is consumed solely in the form of electricity, in amounts depending on the influent character i.e. brackish or seawater. The consumption of energy increases with the level of salinity in the feed water [4]. Low recovery ratio is another environmental problem associated with RO which needs to be addressed, especially for India as an agricultural country. Rejected water with high salinity affects plant growth, resulting in lower crop yields and reduced agricultural production. India harbours many reservoirs of brackish water, such as Chilika Lake in Odisha State which, with an area of 1100 sq. km, is the largest brackish water lake in Asia. Thus there is an acute need of desalination technology, especially in rural areas where power supply is typically poor and communities may depend solely on brackish ground water. According to the Central Ground Water Board of India, ground salinity affects many regions across the country, including Gujarat, Rajasthan, Madhya Pradesh, Odisha, West Bengal, Karnataka and Tamil Nadu, with typical salinity levels in the range of 1500-2500 ppm and sometimes as high as 5000 ppm total dissolved salts [5].

Unreliability of power supply in rural areas is another serious concern. Availability of electricity varies from 8 to 10 hours a day and, even when available, power is highly erratic with crippling voltage fluctuations and frequent outages [6]. A thermal energy-based solution with high recovery ratio is ideal for rural areas, as typically they have biomass in plenty. Solar energy is also abundantly available in India and many other developing countries such as Pakistan, Bangladesh, and Tunisia. In such countries, use of thermal energy in RO-membrane processes has hardly been explored, except through some preheating of feed water to enhance permeability through the membrane [7]. Thermal desalination technologies currently practiced, such as Multiple Effect Distillation (MED), require more energy than membrane processes and consume energy in both electrical and thermal forms [8].

The most prudent use of energy may be achieved by combining thermal sources of energy with membrane RO technology. Very little work has been reported to explore this idea so far. Only a few researchers tried connecting the RO process with solar thermal power plants. One such plant in Egypt in 1981 used the Organic Rankine Cycle (ORC) with Freon-11 as the working fluid [9]. Freon-11, due to its ozone depletion potential, has since been banned under the Montreal protocol. García-Rodríguez and Delgado-Torres performed calculations based on some other organic fluids [10, 11]. Some fluids they considered, like benzene and toluene, are toxic in nature. Besides the choice of working fluid, the concept of coupling the ORC with RO has other design challenges. In such low-temperature applications, heat transfer inefficiencies are highly detrimental. These inefficiencies depend very strongly on the thermodynamic characteristics of the fluid and on the operating conditions. Whereas it is already commonplace to use steam-Rankine cycle power plants to generate electricity for the grid, the possibility of coupling RO-desalination equipment directly with the steam Rankine cycle merits further exploration [12].

In a membrane or RO process, a conventional pressure pump system delivers water at a constant rate irrespective of the concentration level of feed water; whereas in the batch system proposed here, a coupling mechanism gradually increases the pressure, as per the requirement due to the increment in osmotic pressure. The batch system thus provides lower energy consumption, and less reject per unit of produced clean water.

To prove this concept, some initial work has already been done at Aston University (Birmingham, UK) where a linkage- RO machine was designed and tested using air pressure to simulate steam [13, 14]. In this machine, the power piston moves along a horizontal axis and, with the help of the linkage mechanism, transfers power to the water piston moving along a vertical axis. The unit is bulky and complicated to construct due to movement of pistons along these mutually perpendicular axes. To overcome these drawbacks, a new modified design is explored in this paper, which is compact and more feasible in terms of ease of operation and economy. The overall aim of this study is to design a range of versions of the this RO machine suitable for manufacture and use in India and other developing countries, with regard to compactness, cost, availability of materials, performance in terms of low specific energy consumption and high recovery ratio and gain output ratio. The specific objectives are as follows:

1. Design a coupling mechanism with readily available materials and parts, to couple the two pistons, and provide suitably increasing mechanical advantage. Features of a good design include better alignment and balance of forces with compact size.
2. Work out an efficient and easy way to operate a steam-driven thermodynamic cycle that may be applied in a batch process to run a cylinder piston system, taking into account heat transfer aspects.
3. Use only components and materials that are environmentally friendly in terms of inputs and outputs.
4. Achieve high gain output ratio (GOR), and high recovery ratio, with low specific energy consumption (SEC).
5. Achieve economic feasibility, in terms of cost of output product, taking into account both capital and running costs.

In this paper, we present an improved design of the batch-RO machine. First, we describe the overall principle of operation of the machine, which consists of a steam power cycle, a coupling mechanism and a pump-RO cycle. Then, we present the theory relating to each of these three subsystems individually. Based on the theory presented, a parametric model is developed, which is used to work out performance and cost. Practical design constraints are explained, in order to arrive at design options with the help of the model. Thus some preferred designs are presented for outputs in the range suitable for use in rural India.

2. Concept description

The drawbacks of the existing machine stem from its asymmetrical design giving large offset forces, and therefore heavy construction and high capital cost [13, 14]. In the new design proposed here (Fig.1), the power cylinder, coupling mechanism and water cylinder all are co-axial along a vertical axis. This gives symmetry of all moving parts about a common axis. Use of rollers to replace connecting rods also helps to make the machine more compact. Thus the line of action and reaction of forces, and the centre of gravity of the whole unit, lie on the same vertical axis such that offset forces are eliminated. Steam from a solar panel or boiler is sent to a power cylinder through a steam jacket that surrounds the cylinder shell. The steam jacket helps to maintain high temperature inside the cylinder to enable an isothermal Rankine cycle (Fig. 2). Isothermal operation improves efficiency, and is achievable in a slow process like this one where cycle time is generally greater than 1 minute. To enhance the heat transfer from cylinder walls to steam, the use of fins inside the cylinder has been explored. This leads to reduced heat flux requirement by a margin of about 40% (Fig. 3). Steam pressure during its expansion stroke pushes the power piston in the forward direction. Movement of the power piston actuates the coupling mechanism (Fig.4) that transfers force to the water piston in the water cylinder with gradually increasing mechanical advantage. Different positions and movements of the power and water piston according to angles $\propto_p$ and $\propto_w$ are illustrated in Fig.4. The water piston pressurizes saline feed water fed to a RO module, which permits only clean water to pass through, mostly excluding salt and other impurities.

After performing the expansion stroke, the piston descends under gravity due to the weight of the moving parts. During this down stroke, another batch of saline water is fed to the water cylinder, and then the process is repeated. After completion of each expansion stroke, low-pressure steam (along with an amount of condensate formed in the power cylinder and jacket) escapes through the exhaust valve, to be re-circulated through a heat exchanger and condenser and again pumped to the solar panel or boiler. At the start of operation, the flow rate of condensate may be higher, but as the steady state is reached it will decrease. In the ideal situation the condensate amount should be the least possible and either is drained out or pumped to the solar/boiler feed line. Clean water received from the RO module is routed through the condenser to provide the necessary cooling.

3. Thermodynamics and heat transfer aspects of the steam power cycle

The detailed operational sequence of the steam system is as follows:

1. Dry saturated steam from the biomass boiler or solar panel is injected into the power cylinder through an annular space between the cylinder wall and jacket (refer to Fig. 1). This helps to reduce loss by condensation in the engine cylinder because of: (i) difference of temperature between the steam and metal of cylinder wall and (ii) increase in volume during slow expansion process. Direct feed of steam into the cylinder and mixed feed may also be carried out as required [15]. This will give several possibilities of experimentation to determine an optimum value of cut off under varying input steam parameters including pressure, temperature and flow rate.

2. Supply of steam continues till cut off point (refer to Fig. 5). The law according to which the pressure of steam diminishes during its expansion depends on the conditions under which the expansion takes place, and on the initial state of the steam. Our process, being very slow, is more likely to behave as isothermal, instead of isentropic or polytropic as in the case of high-speed engines. For elementary purposes, then, it is sufficient to assume the simple approximate law that the absolute pressure will vary inversely with volume according to the ideal gas law [15].

3. As the piston arrives at the end of its stroke and the expansion is finished, the communication with the receiver or condenser is opened and the steam along with condensate escapes at the lower pressure. The work done in principal may be increased in two ways: (i) by maximum condensation of steam at the end of stroke and (ii) by increasing the expansion during the stroke [16].

The effect of the jacket is therefore to reduce the extent of the heat interchange between the steam and the walls, and, as a consequence, to reduce also the weight of water to be afterwards re-evaporated at the expense of heat from the cylinder walls. Thus, with a jacket, a smaller proportion of the heat of the steam is wasted in merely heating the walls, and a larger proportion is employed in the performance of useful work than when no jacket is used. All heat transmitted through the cylinder walls from the jacket is accompanied by a corresponding loss due to condensation in the jacket.

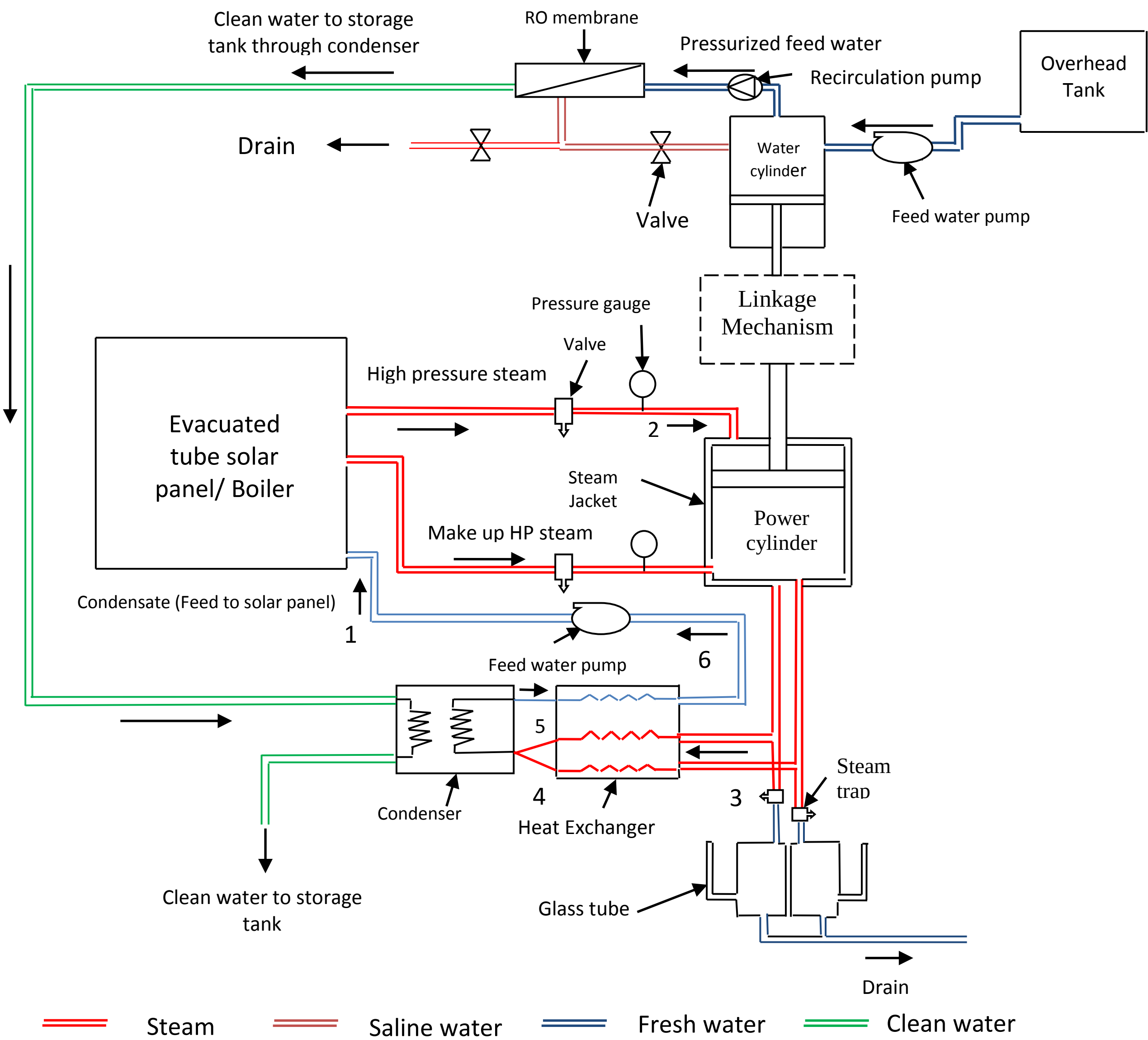


Fig. 1. Schematic view of the proposed steam-driven RO desalination plant with steam raised in a solar or biomass boiler.

3.1 Thermodynamic cycle

The temperature-entropy diagram of a hypothetical isothermal Rankine cycle is given in Fig.2. The work is delivered during the expansion stroke of the cycle, represented by process 2-3. In most steam engines, the expansion stroke is considered to be isentropic because of high-speed operation. In our case, as the process is very slow (typically taking more than sixty seconds to complete), an isothermal expansion is the preferred concept.

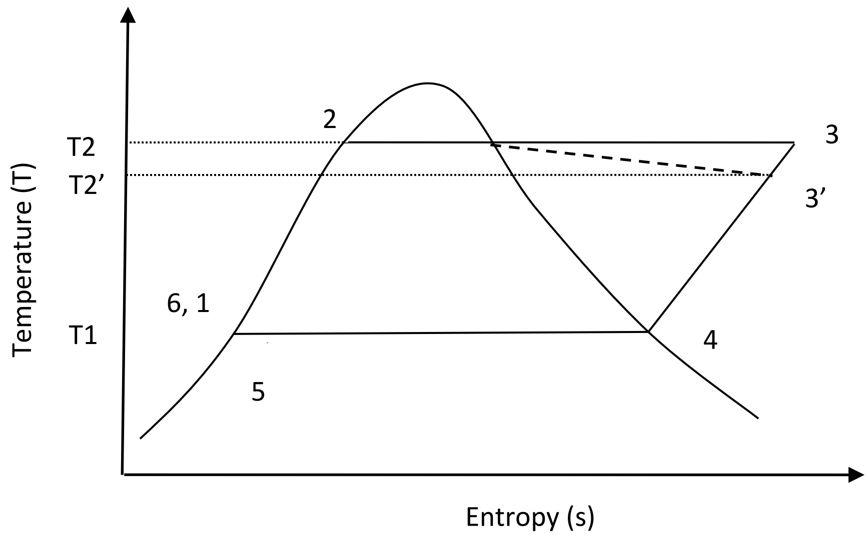


Fig.2. T-s diagram of for the concept of the isothermal Rankine cycle

The efficiency of the cycle is the ratio of heat energy utilized to that supplied:

$$\eta = \frac{q_{in} - q_{out}}{q_{in}} \tag{1}$$

In practice, the process may not be absolutely isothermal and temperature may fall at the end of the expansion stroke to point 3' instead of 3 (Fig.2). For ease of calculation it is nonetheless assumed to be isothermal. A similar study was performed using a hot oil jacket surrounding the power cylinder leading to the following expression for efficiency [17].

$$\eta = 1 - \frac{\frac{T_c}{T_h}}{1 - \alpha\left[\left\{\frac{T_c}{T_h}\right\} - 1 - ln\left\{\frac{T_c}{T_h}\right\}\right]} \tag{2}$$

Where T_h and T_c are the temperatures of input steam at the cylinder and of the condenser respectively, and α is a constant represented by following equation:

$$\alpha = \frac{T_c(C_{pf} - C_{pg})}{h_{fg}} \tag{3}$$

Where C_{pf} and C_{pg} are specific heat capacities in liquid and gas states. The theoretical efficiency of the cycle approaches the Carnot efficiency for our operating range of pressure and temperature.

3.2 Heat transfer

The cylinder-jacket assembly works like a shell-and-tube type heat exchanger where heat is to be transferred from the jacket into the cylinder to maintain the high temperature of steam inside the cylinder during the slow expansion process. Use of a steam jacket in place of other heat transfer fluids (such as thermic oil) gives better control and better heat performance, as latent heat is utilized. Steam in the jacket condenses and releases latent heat into the cylinder as it changes in phase from vapour to liquid. This way a large amount of heat energy is released as compared to thermic oil that would only supply sensible heat. Furthermore, the amount of heat transferred may easily be calculated monitoring the flow rate of condensate in glass tube indicators and may be regulated with the help valves mounted in cylinder and jacket line separately.

Use of fins inside the power cylinder has also been explored for increased heat transfer; however this leads to few complications associated with sealing of the piston. It is seen that heat flux requirement reduces by about 40% with the use of 40 square fins of 10 mm size along height of the cylinder (Fig. 3). This amount of heat is transferred from steam jacket via convection and radiation. A heat flux of 150-200 W/m^2 in a slow process may easily be transferred by convection and radiation together. The convective component is:

$$q_{convective} = hA\Delta T \tag{4}$$

where h is heat transfer coefficient, A is heat transfer surface area and ΔT is temperature difference between cylinder wall and steam inside the cylinder.

Convective heat transfer coefficient approaches a high value in the case of phase change [18]. It may go up to 5000 W/m^2 K for one side phase change and even higher if phase change occurs at both sides of the cylinder wall. This is the advantage of using steam as the heating fluid, as it delivers latent heat on phase change. The radiative component of heat transfer is:

$$q_{radiation} = F_{g12}\sigma A({T_w}^4 - {T_s}^4) \tag{5}$$

$F_{g12}$is a grey body factor which depends on emissivity of the wall and steam, and their temperatures, σ is the Stefan-Boltzmann constant, and T_w, T_s are the absolute temperatures of wall and steam respectively.

$$F_{g12} = \frac{1}{\frac{1}{\varepsilon_1}-\frac{1}{\varepsilon_2}-1}, \tag{6}$$

Where ε_1 andε_2 are the emissivities of wall and steam respectively.

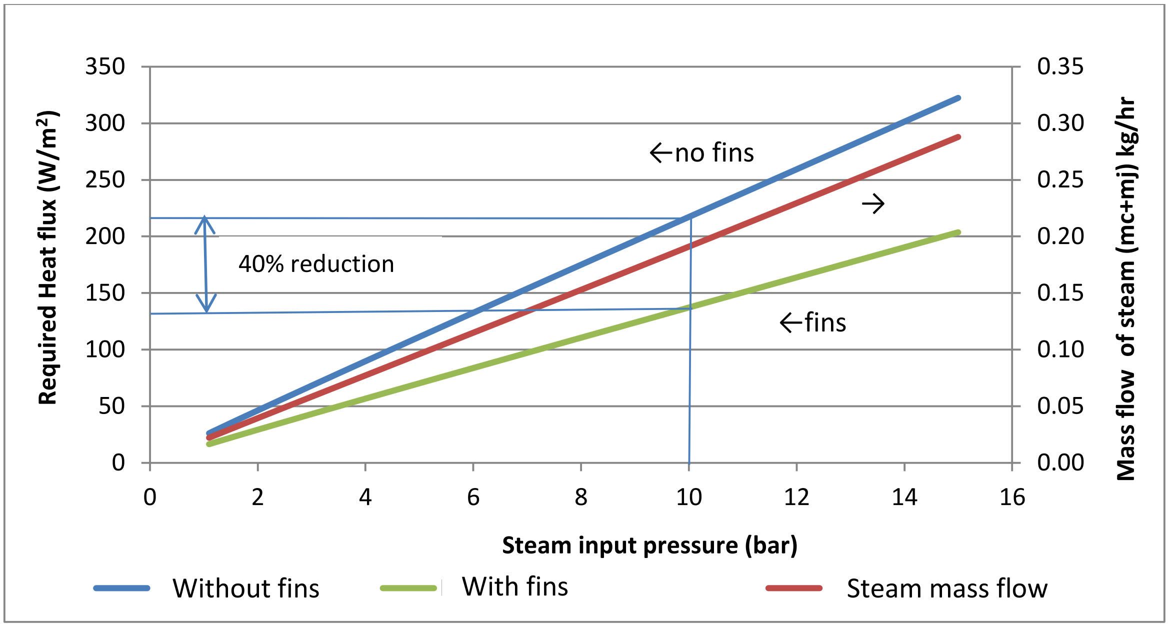


Fig. 3. Heat flux required with and without fins for the isothermal process, at different steam input pressures, showing a 40% reduction through use of fins

4. Analysis of coupling mechanism and RO system

The coupling mechanism uses a right-angled triangular crank plate with dimensions of its two working sides denoted as L_p and L_w towards power piston and water piston respectively (Fig 4). Selection of operating angles is chosen to vary from 75^o to 5^o at the power side and from 15^o to 85^o at the waterside. Extreme horizontal and vertical angles such as 0^o and 90^o are avoided to prevent the mechanism from jamming. These angles are measured from a common horizontal axis passing through both symmetric and aligned pivots. A symmetrical design was preferred to prevent side forces that would necessitate a heavy frame and linear bearings.

The pump pressure must be higher than the osmotic pressure in order to force saline water through the membrane. The flow rate is proportional to the difference between the two pressures. When they are equal, water does not flow through the membrane, and if the pump pressure is lower than the osmotic pressure, permeate water may even flow back towards the concentrated salt water. The osmotic pressure of the feed water (P_{osmf}) due to salinity may be determined using the Van't Hoff formula [19] as:

$$P_{osmf} = cRT \tag{7}$$

where c is molar concentration, R is gas constant and T is temperature in K.

This study has been carried out for brackish water of salinity level of around 4000 ppm containing mostly NaCl. Osmotic pressure is thus about 3 bar, assuming feed water at ambient temperature of 32^{o}C.

The amount of energy required to produce desalinated water from saline water is given by integration of the van't Hoff law:

$$E = VP_{osmf}\frac{1}{r}ln\frac{1}{(1-r)} \tag{8}$$

Where, V is the volume of fresh water produced and r is recovery ratio, which is the ratio of the desalinated water volume to the feed water volume.

Stroke lengths and corresponding volumes available in the power cylinder and water cylinder are determined using simple trigonometric and geometric relations as:

Power stroke length $L_p = R_p(\sin\alpha_{p1} - \sin\propto_{p_2})$; Volume = $A_p \times L_p$ (9)

Water stroke length $L_w = R_w(\sin\alpha_{w1} - \sin\propto_{w2})$; Volume = $A_w \times L_w$ (10)

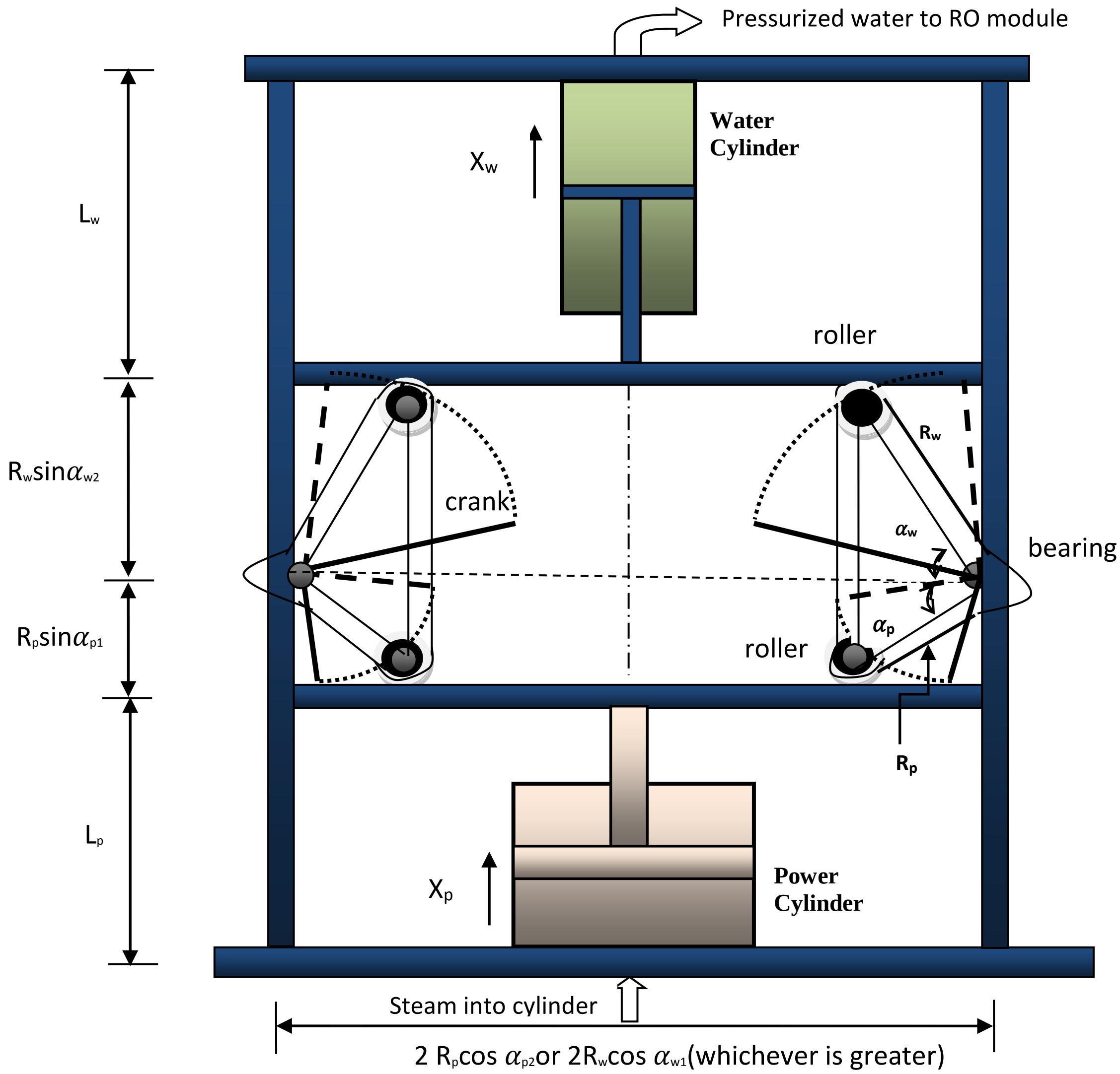
For modelling purposes, a series of small steps in the crank position are considered. Corresponding piston positions and cylinder volumes are determined. Steam pressure in the power cylinder is allowed to build up till cut-off point; then the steam expands isothermally to perform the power stroke following the relation PV = constant. The mass of steam required in the power cylinder to complete one cycle is determined by multiplying the density of steam at the corresponding input pressure with the volume supplied till cut-off point.

The volume in water cylinder consists of stroke volume and a minimum amount of water already present in RO module as V_{min}. We assume V_{min} to be 1 litre. The osmotic pressure, initially given by equation (7), keeps on increasing with incremental decrease in volume and corresponding increase in salt concentration. At each step of the calculation, the direct force on each piston is calculated as the product of pressure and the respective piston area.

The mechanical advantage achieved with this design is the ratio of the horizontal distances from the pivot to crank end of power piston and water piston respectively. This mechanical advantage leads the water piston to work against increasing osmotic pressure at the water piston. At any instant it is given as:

$$\text{Mechanical Advantage (MA)} = \frac{\mathrm{R_p}\ \cos\alpha_p}{\mathrm{R_w}\ \cos\alpha_w} \tag{11}$$

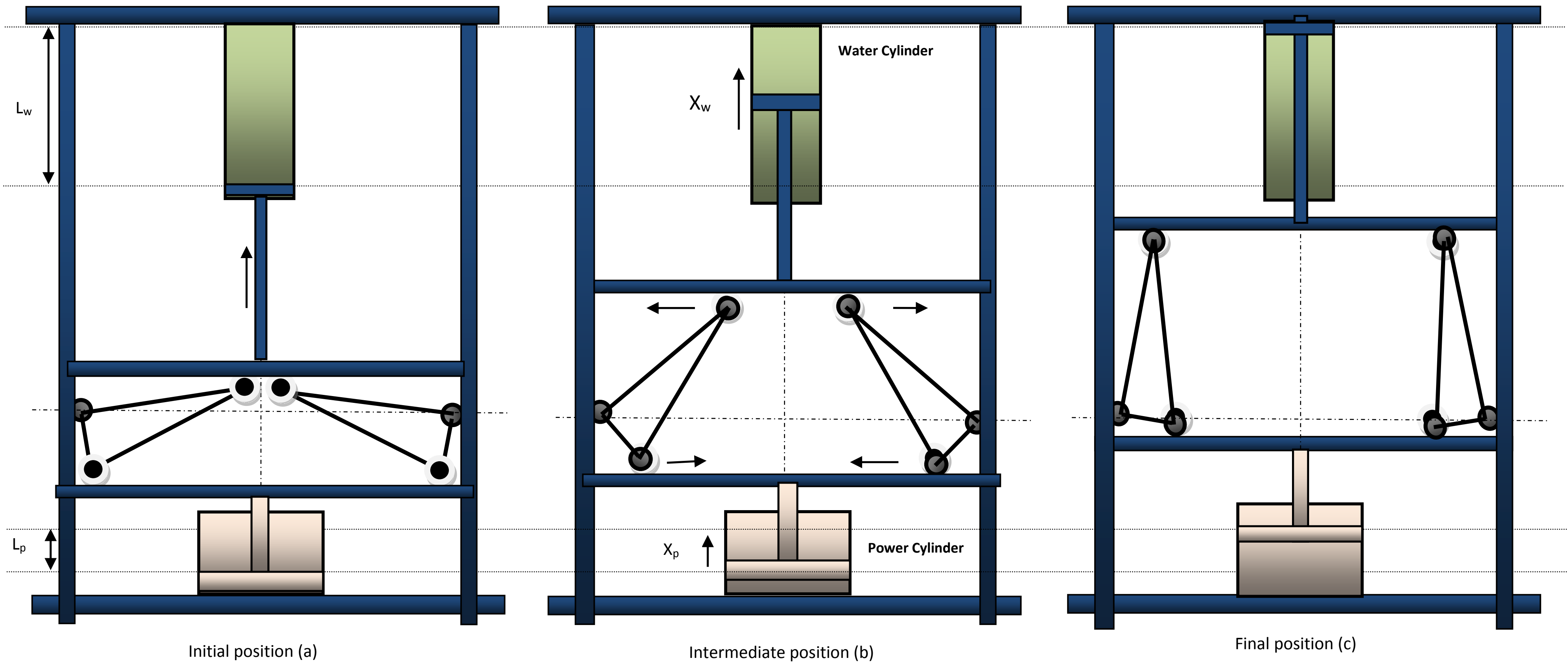
Actual instantaneous force at the water piston is the product of force at power piston and mechanical advantage. Thus the net driving pressure at the water piston is calculated by subtracting direct pressure from actual pressure at the water piston.



$$\propto_p = 75 - 5^0\ ;\ \propto_w = 15 - 85^0;\ R_p\ = 0.17\ m\ ;R_w\ = 0.5\ m$$

$$Power\ cylinder\ stroke\ length\ X_p\ = 0.15\ m\ ;\ Water\ cylinder\ stroke\ length\ X_w\ = 0.37\ m$$

Fig. 4. Coupling mechanism using twin cranks with rollers, designed to achieve increasing mechanical advantage throughout the cycle of operation. Initial position of crank is represented by bold lines and final position by dotted lines. Length of the unit will be 2 $R_p \cos \alpha_{p1}$ or 2 $R_w \cos \alpha_{w1}$ (whichever is greater) plus some clearance distance. Height may also be worked out as mentioned in figure.



$\alpha_p = 75^0$,steam injected $\alpha_p = 47^0$, cut off, steam injection stopped $\alpha_p = 5^0$, cycle completes

Fig. 5. Different positions of power piston and water piston with variation of angles $\propto_p$ and $\propto_w$.The figure illustrates two extreme positions as initial and final positions and one intermediate position. At the start of cycle, steam is allowed to enter into power cylinder till cut of point as referred to intermediate position. At cut off, fresh steam supply stops and steam in the cylinder expands to deliver the work. At the end of expansion, the piston comes to rest as referred to final position and cycle completes. Steam at low pressure is allowed to exhaust. Both the pistons fall back to their original position to repeat the cycle again. Water cylinder is again filled for next cycle. The process in accordance with steam is described in detail in thermodynamic and heat transfer aspect section of this paper ahead.

The above equations were incorporated into a spreadsheet model, which was used to adjust the dimensions of the crank plate and diameters of the cylinders, so as to minimize the overall size of the machine. It can be visualized from the figure above that stroke length and cylinder volumes at delivery end (water cylinder side) and receiver end (power cylinder side) depend on selection of lengths of cranks and angles. The displacements of the power and water pistons, with variation of angular position of power crank, are shown in Fig. 6.

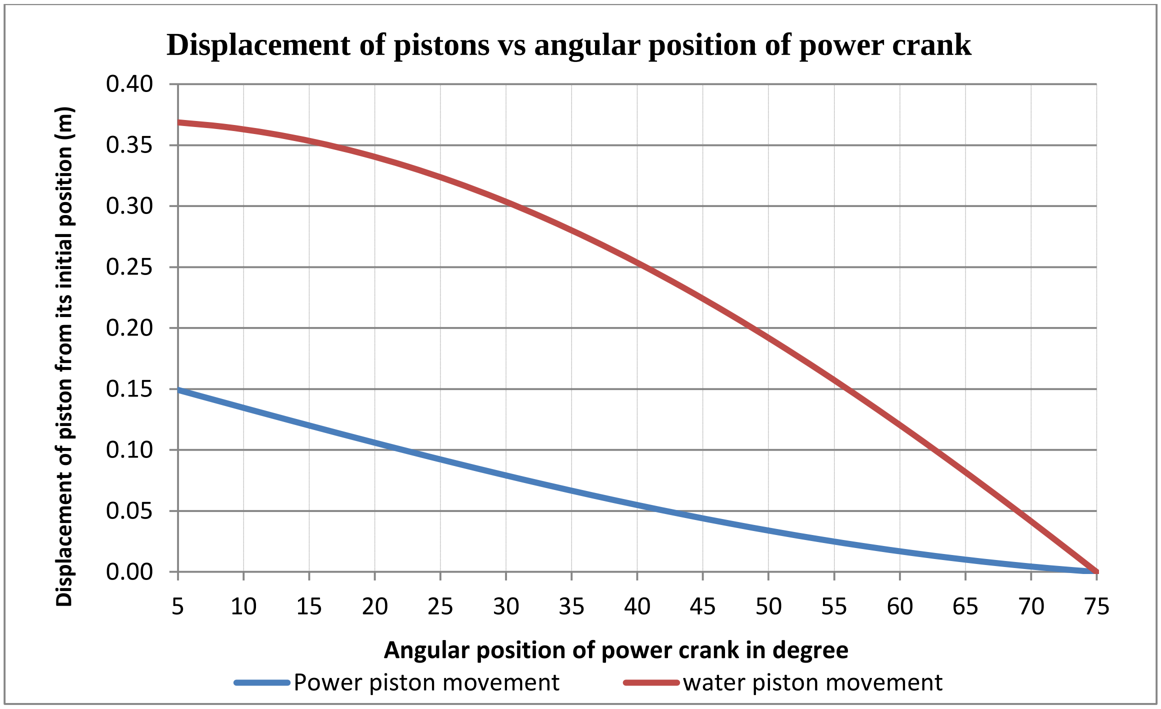


Fig. 6. Displacement of power and water pistons with variation of angular position of power crank for an example case (*d_p= 305 mm, d_w=152 mm, R_p= 170 mm, R_w= 210 mm)*

The magnitude of force that is transferred from power piston to water piston is a function of mechanical advantage. The difference in this transferred force and the force corresponding to the osmotic pressure in the water cylinder is the net driving force, responsible for driving the saline water against RO membrane. Ideally net driving force should be constant, but this is not achieved in practice. Fig. 7 demonstrates stage wise: the pressure in the power cylinder and water cylinder, mechanical advantage, and net driving pressure available at water piston. These parameters have been calculated at different incremental angular positions, which are selected as $\propto_p = 75 - 5^0$ for power cylinder side crank, and $\propto_w = 15 - 85^0$ for water piston side crank measured from horizontal axis passing through the pivot.

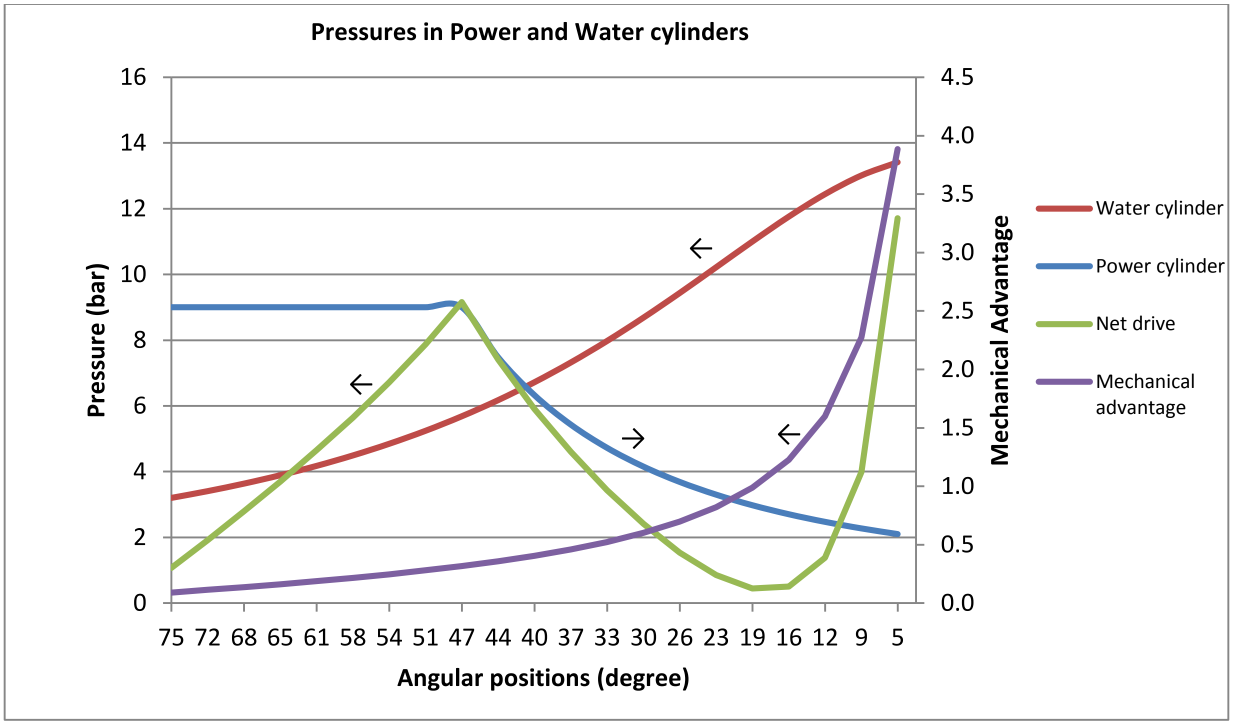


Fig. 7.Pressure in power and water cylinder, mechanical advantage and net driving pressure for an example case (*d_p= 305 mm, d_w=152 mm, R_p= 170 mm, R_w= 210 mm)*

Osmotic pressure continuously increases during the cycle as the concentration salt concentration increases. The time elapsed in the RO filtration is flow rate of treated water from the membrane against volume passed. Flow rate from the membrane is given by

$$Q = P_{net} \times A_{mem} \times s \tag{12}$$

Where P_{net} is the net driving force at the water piston, A_{mem} is the area of RO membrane (which is taken as 2.6 m^2) and s is the membrane permeability (which is taken as 2.2 x 10^{-11}m/sPa for membrane type XLE- 2540 for design and modelling purpose [20]). Adding up all the incremental times, the total cycle time is calculated.

5. Results and discussion

During the course of design and modelling, several options were analyzed with varying component sizes and operating pressures. The main size parameters which may vary in designing the machine are as follows:

1. Power cylinder diameter, d_p
2. Water cylinder diameter, d_w
3. Length of crank at power cylinder side, L_p

4. Length of crank at water cylinder side, L_w

Modelling and analysis have been performed based on readily available materials and components from the market in India, to work out the capital cost. Specifications and dimensions of such components and other input parameters are summarized in Table 1.

Table 1. Selection of components for modelling and parametric analysis

Parameter	**Description**
Power cylinder Diameter d_p	254/ 305/ 406/ 610 mm (10"/ 12"/ 16"/ 24")
Water cylinder diameter d_w	102/152 mm (4"/ 6")
RO membrane element type	XLE- 2540 (Filmtec)
RO membrane area (A_{mem})	2.6 m^2
RO membrane Permeability (S)	2.2×10^{-11} m/s.Pa
RO membrane salt transfer coefficient (B)	2.9×10^{-7} m/s
Feed water salinity	4000 ppm
Input steam pressure range	6- 15 bar (abs)

Each combination of these parameters, together with varying input steam pressure and temperature, provided a corresponding set of output parameters including:

1. Stroke lengths (mm)
2. Force on the pistons (kN)
3. Net driving pressure at the membrane (bar)
4. Cycle time (seconds)
5. Production output of freshwater (l/min)
6. Output salinity (ppm)

The results, summarized in Table 2, were assessed with regard to the following desirable qualities:

1. Design should be compact. This reduces the capital cost of machine.

2. Appropriate variation of mechanical advantage to minimize change in net driving pressure.
3. Forces on pistons should not be too high to avoid high capital and maintenance costs.
4. Net drive pressure above 30-35 bar should be avoided, as heavy duty piping and connections will be needed. Lower pressure also increases the life of the membrane.
5. Designs with less cycle time and higher are preferred, but not at the cost of GOR and specific energy consumption
6. The unit should tolerate a range of pressure, meaning that if it gives best performance at certain pressure P, it should all perform well over a range say P$\pm$ 1bar. This will ensure robustness in operation.

On the basis of given criteria, three designs are selected after critical examination marked as (A), (B) and (C) in Table 2. These designs were analyzed in terms of output and specific steam consumption.

Fig. 8 shows the quantity of treated water produced in m^3 per day with three selected designs at different operating pressures. It is evident that design (C) gives best results in terms of total water production per day. In terms of robustness it is also tolerant of a range of operating pressures.

Further investigations to output in relation to steam consumption were carried out, as represented in Fig. 9. Design (A) produces up to 4 m^3/day of water at 12–13 bar pressure. Design (B) also produces about 4–5 m^3/day using pressure range of about 13–14 bar. At the same pressure range, design (C) delivers about 7 m^3/day at lower specific water output, per unit of steam. In general, increase in steam pressure favours greater water output per day from a given design, but at the expense of more steam consumed per m^3 of water produced.

These results correspond to low values of specific energy consumption based on thermal energy supplied (SEC_{th}). For example, Design A with steam supplied at 9 bar gives SEC_{th} of 4.4 kWh/m^3; and Design C at 7 bar gives SEC_{th} of 5.5 kWh/m^3. These values compare very favourably to thermally driven processes based on distillation, where SEC_{th} > 50 kWh/m^3 could typically be expected even with multiple effects.

6. Economic feasibility

The capital and running costs of the machine have been analyzed to calculate product water cost in the case of a biomass boiler being used to generate steam. The analysis has been done for the most promising design (C) having the specifications of: power piston diameter 305 mm, water piston diameter 152 mm, crank length at power piston side 170 mm and crank length at water piston side 210 mm and steam operating pressure of 11 bar (abs). This is found to be the most compact design possible with the available sizes of pipes for use as cylinders, while operating within the normal pressure range of 7−11 bar, thus allowing a range of standard equipment to be used. Use of standard sizes and operating pressures helps keep capital cost low.

Table 3 gives the cost breakdown by component. Strength calculations have been carried out to determine the dimensions of the different components including steel frame and rollers. The required strength for different components has been determined based on the maximum loads of 73 kN and 106 kN at the power and water pistons respectively. A minimum safety factor of at least 5 has been used throughout.

The repayment instalment R to amortise the capital cost Cc with annual interest rate *i* is calculated for life span of *n* years using following equation [21].

$$R = Cc\left[\frac{i(1+i)^n}{(1+i)^n - 1}\right] \quad \text{or} \qquad R = Cc \times 0.16274 \tag{13}$$

Assuming a rate of interest as 10% per annum and expected life of plant 10 years, repayment instalment comes out to be INR 47553/year (INR: Indian Rupee). Considering 300 working days in a year, the repayment instalment is INR 159/day. Maintenance cost has been taken as 20% of fixed cost and is calculated to be INR 195/day.

Table 2.Design combinations with varying system parameters (preferred designs highlighted in bold).

Power cylinder diameter, d_p (mm)	Water cylinder diameter, d_w (mm)	Power cylinder side crank radius, R_p (mm)	Water cylinder side crank radius, R_w(mm)	Observations							Max force on piston (kN)		Cycle time (s)	Output (L/min)	Steam consumption (kg/hr)	Gain output ratio GOR
				Stroke length (mm)		Size of Machine		Mechanical Advantage		Maximum Net Drive Pressure (bar)						
				Power piston	Water piston	Length (mm)	Height (mm)	Min	Max		Power	Water				
254 (10 inch)	102 (4 inch)	170	540	150	400	1040	1250	0.08	3.6	Low yield, big size – not preferred						
Operating input steam pressures: (bar absolute)				9						12	41	20	209	0.92	0.34	162
				11						22	51	29	95	2.05	0.95	129
				13						33	61	37	66	2.93	1.56	113
				15						43	71	46	51	3.77	2.34	97
254 (10 inch)	**152 (6 inch)**	**170**	**240**	**150**	**180**	**460**	**730**	**0.19**	**8.1**	**(A) preferred**						
Operating input steam pressures: (Bar absolute)				9						12	41	45	211	0.92	0.34	162
				11						22	51	64	95	2.05	0.95	129
				13						33	61	84	66	2.93	1.56	113
				15						43	71	103	51	3.77	2.34	97
305 (12 inch)	102 (4 inch)	120	470	100	350	910	1080	0.07	2.86	Low yield, big size – not preferred						
Operating input steam pressures: (bar absolute)				8						10	51	18	167	1.01	0.43	141
				9						16	58	23	101	1.68	0.79	128
				11						28	73	33	63	2.71	1.48	110
				13						41	88	43	46	3.68	2.37	93
				15						53	102	52	37	4.64	3.46	80
305 (12 inch)	102 (4 inch)	170	470	150	350	910	1130	0.1	4.11	Low yield, big size – not preferred						
Operating input steam pressures: (bar absolute)				7						12	44	19	95	1.78	0.87	123
				8						20	51	26	65	2.6	1.44	108
				9						29	58	33	51	3.33	2.08	96
				11						46	73	47	36	4.74	3.55	80
				13						64	88	61	28	6.12	5.45	67

Power cylinder diameter, d_p (mm)	Water cylinder diameter d_w(mm)	Power cylinder side crank radius, R_p(mm)	Water cylinder side crank radius, R_w(mm)	Observations							Max force on piston (kN)		Cycle time (s)	Output (L/min)	Steam consumption (kg/ hr)	Gain output ratio GOR
				Stroke length (mm)		Size of Machine		Mechanical Advantage		Maximum Net Drive Pressure (bar)						
				Power piston	Water piston	Length (mm)	Height (mm)	Min	Max		Power	Water				
305 (12 inch)	**152 (6 inch)**	**120**	**210**	**100**	**150**	**400**	**630**	**0.15**	**6.5**	**(B) preferred**						
Operating input steam pressures: (bar absolute)				8						11	51	41	161	1.05	0.42	150
				9						17	48	52	99	1.7	0.74	138
				11						29	73	74	62	2.73	1.45	113
				13						41	88	96	45	3.71	2.34	95
				15						53	102	118	36	4.67	3.42	82
305 (12 inch)	**152 (6 inch)**	**170**	**210**	**150**	**150**	**400**	**680**	**0.22**	**9.3**	**(C) preferred**						
Operating input steam pressures: (bar absolute)				7						12	44	43	93	1.81	0.85	128
				8						21	51	59	64	2.63	1.42	111
				9						29	58	75	50	3.36	2.00	101
				11						47	73	106	35	4.77	3.53	81
				13						64	88	138	27	6.16	5.38	69
406 (16 inch)	152 (6 inch)	230	290	200	210	560	870	0.21	9.07	High force on piston and membrane – not preferred						
Operating input steam pressures: (bar absolute)				6						11	65	47	77	3.05	2.28	80
				8						41	90	102	40	5.84	5.56	63
				10						71	117	157	28	8.29	9.68	51
406 (16 inch)	152 (6 inch)	230	350	200	260	680	980	0.18	7.51	High force on piston and membrane – not preferred						
Operating input steam pressures: (bar absolute)				6						19	65	67	101	2.79	1.73	97
				8						29	90	85	65	4.34	3.42	76
				10						54	117	130	44	6.43	6.23	62
610 (18 inch)	152 (6 inch)	340	410	300	300	790	119	0.22	9.48	High force on piston and membrane – not preferred						
Operating input steam pressures: (bar absolute)				6						41	146	112	32	10.4	17.0	37
				7						76	175	176	25	13.4	26.3	31

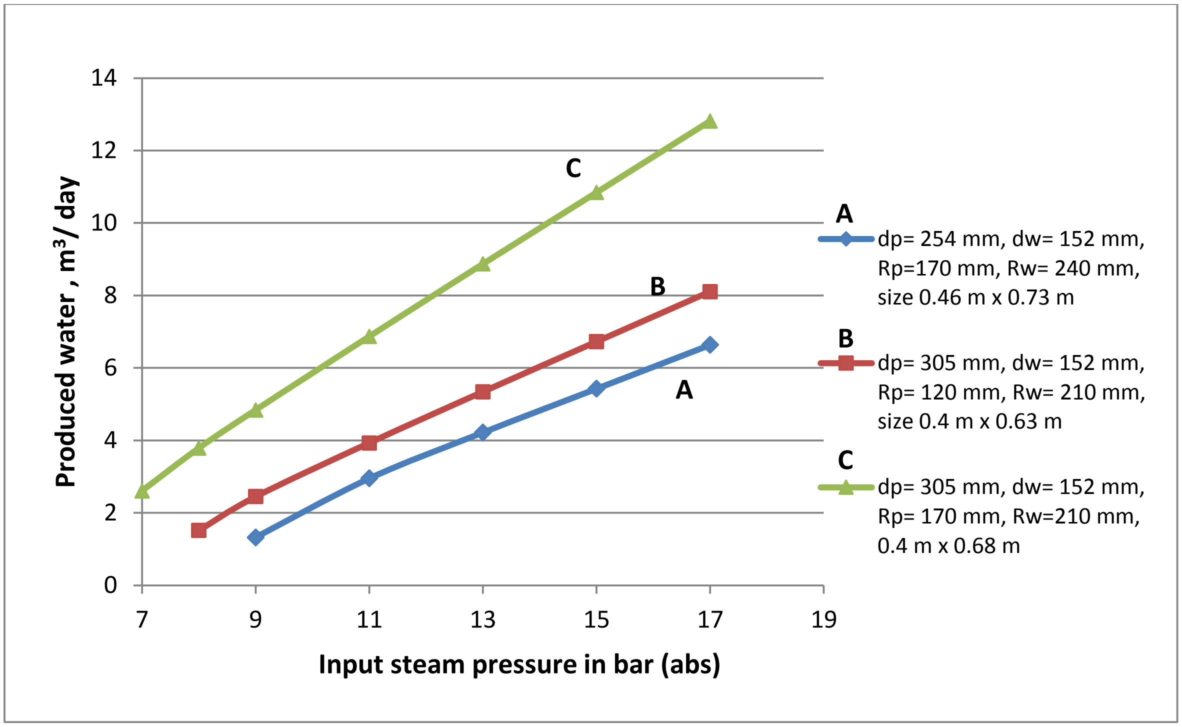


Fig. 8. Volume of water produced per day with three preferred designs, with different input steam pressures.

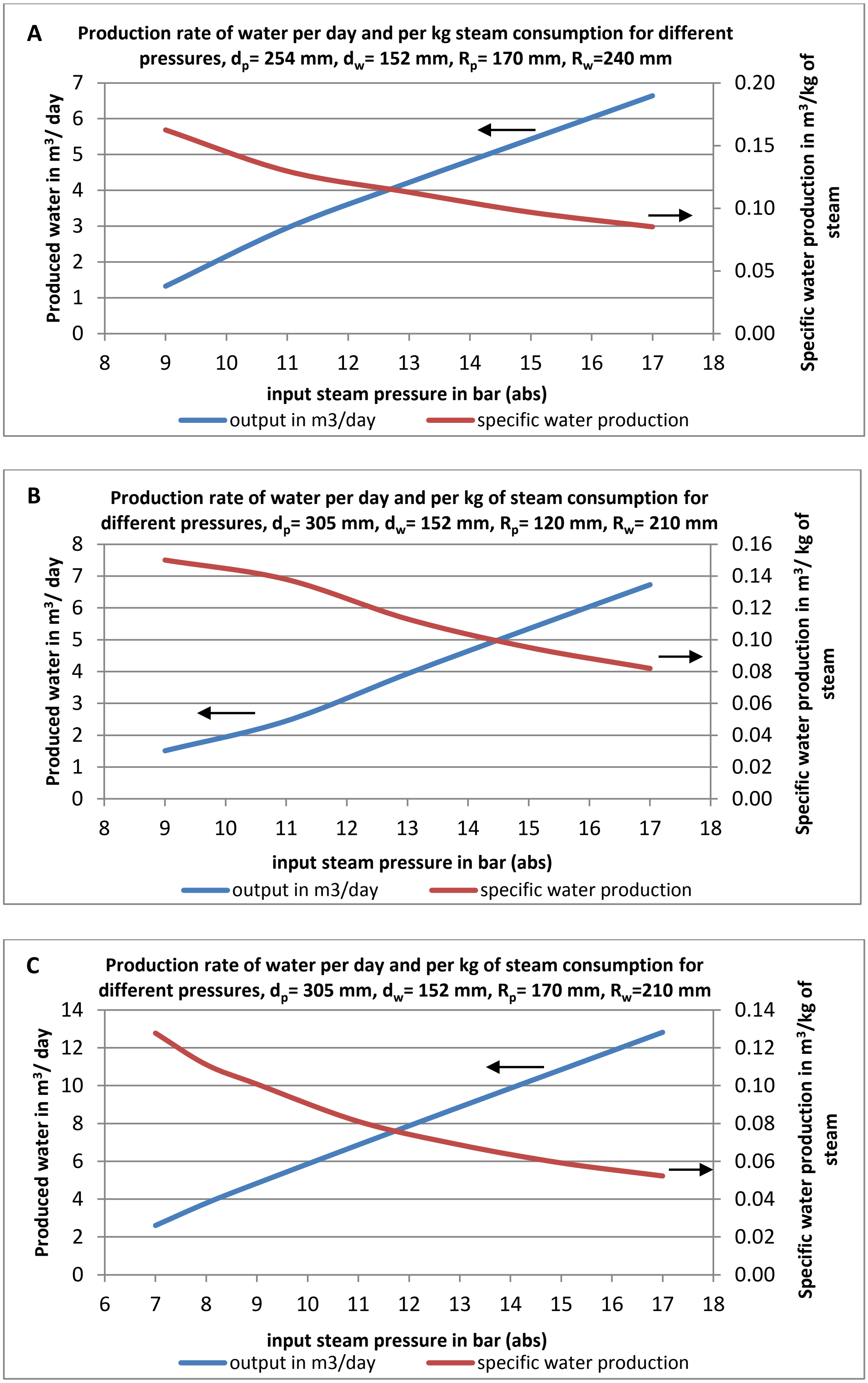


Fig. 9. Volume of water produced per day and per kg of steam consumed, at different input steam pressures, for preferred designs A, B and C.

Table 3.Cost model and strength requirements for main components (Design C)

Part	**Size (mm)**	**Max load or stress (σ)**	**Material**	**Cost (INR)**	**Quantity**	**Total cost (INR)**	**Supplier**
Vertical columns, both sides (to support push rail and pivots)	100 × 100 square channel × 5 thick × 1500 long	$\sigma = 5.4\times 10^4$ kN/m^2	Mild steel (EN10025 S275JR)	2000	2	4000	Jasubhai & Company, Ahmedabad, India
Vertical columns, front and rear (for extra support to water cylinder)	100 × 100 square channel × 5 thick × 1500 long	$\sigma = 5.4\times 10^4$ kN/m^2	Mild steel (EN10025 S275JR)	2000	2	4000	Jasubhai & Company, Ahmedabad, India
Horizontal beam (joining the side columns at top and bottom)	100 × 100 square channel × 5 thick × 1300 long	$\sigma = 5.4\times 10^3$ kN/m^2	Mild steel (EN10025 S275JR)	2000	2	4000	Jasubhai & Company, Ahmedabad, India
Horizontal beam (joining the front and the back column)	100 × 100 square channel × 5 thick × 500 long	$\sigma = 5.4\times 10^3$ kN/m^2	Mild steel (EN10025 S275JR)	1500	2	3000	Jasubhai & Company, Ahmedabad, India
Horizontal push rail	75 mm C- section × 5 thick × 1160 long	$\sigma = 1.2\times 10^3$ kN/m^2	Mild steel (EN10025 S275JR)	1500	2	3000	Jasubhai & Company, Ahmedabad, India
Forklift truck roller	180 ×65; Ø 25 HL 75	53 kN load	Polyurethane	5000	4	20000	TENTE Castors limited, Ireland
Crank plate	500× 170 × 530		Mild steel (EN10025, S275JR)	3000	2	6000	Jasubhai & Company, Ahmedabad, India
Bearing	Ø 25	53 kN load		1500	6	9000	SKF, Ahmedabad, India
Bearing block	For bearing	53 kN load		1500	6	9000	SKF, Ahmedabad, India
Coupler bolt	Ø 30×70	$\sigma = 7.4 \times 10^4$ kN/m^2	Stainless steel, S416	300	8	2400	Shreeji Engineering, Ahmedabad, India

Steam powered cylinder-piston arrangement with vertical square fins projected into the cylinder covered with steam jacket over the cylinder shell	Ø 305 x 200	$\sigma = 1\times 10^3$ kN/ m2	cast iron, grey	30000	1	30000	Shreeji Engineering, Ahmedabad, India
Power piston sealing with grooves	UHS, 230 × 250 × 12	$\sigma = 3.8 \times 10^3$ kN/ m^2	Polyurethane	10000	1	10000	Hiflon polymers idustries, Ahmedabad, India
Water cylinder- piston arrangement	Ø 152× 200		Cast iron, grey	15000	1	15000	To be fabricated or forged
RO module			XLE- 2540	30000		30000	Lenntech
Steam valve	Ø 25		Stainless steel, S 416	2000	2	4000	Shreeji Engineering, Ahmedabad, India
Steam trap	Ø 25		Cast iron	2000	2	4000	Shreeji Engineering, Ahmedabad, India
Pipe line	Ø 25		Galvanized iron	500	20 m	10000	Shreeji Engineering, Ahmedabad, India
Insulation material				500	20 kg	10000	Shree firepack safety pvt. limited, Ahmedabad, India
Solar panel and accessories	Parabolic trough solar energy collector or evacuated tube solar panel		Standard as Company specified	50000	1	50000	Benchmark Engineers And Consultants Coimbatore - 641012, Tamil Nadu, India
Biomass boiler and accessories	Dry saturated steam: 3-4 kg/hr, 175-200ºC, 10-15 bar (g)		Standard as Company specified	60000	1	60000	Balkrishna boilers private limited, Ahmadabad, india
Pressure gauge	1-20 bar			400	6	2400	
Thermocouple	100-250ºC			400	6	2400	
					TOTAL	292200	

Based on biomass consumption of 5 kg/hr, the fuel cost is INR 600 per day @ INR 5 per kg. The boiler considered produces 20 kg/hr of steam, enough to feed to at least five RO machines simultaneously. So per machine, fuel cost may be taken as INR 120/day. A 30 W feed pump and 300 W air blower for the boiler, and 60 W recirculation pump for the RO, consume 3 kWh of electricity per machine per day. The price of electricity is taken as INR 8 per kWh, resulting in a cost of INR 24/day. The biomass-based system is intended for rural areas which may be run on a cooperative basis in shifts by local villagers, so labour cost has not been included. Total daily cost, as the sum of these costs, thus comes to INR 498/day (Table 4). With production from each machine of 7 m^3/ day, the cost of treated water will thus be INR 71/m^3. This cost will further decrease if the system runs for more than 300 days in the year.

Table 4: Daily costs (INR) for machine with 7 m^3/day output (Design C).

Repayment instalment	159
Maintenance cost	195
Fuel cost	120
Electricity	24
TOTAL	498

Fuel cost in this design contributes only 24% of the total cost and is the second smallest of the four major contributions (Fig. 10). This may be reduced further if steam is made available as a by-product (for example, in the form of waste steam from industry, or by using exhaust gas from a small diesel engine to generate steam via a heat exchanger). Then the primary steam may be obtained effectively free, and the cost would reduce to about INR 60/m^3 (less than 1 US dollar/m^3).

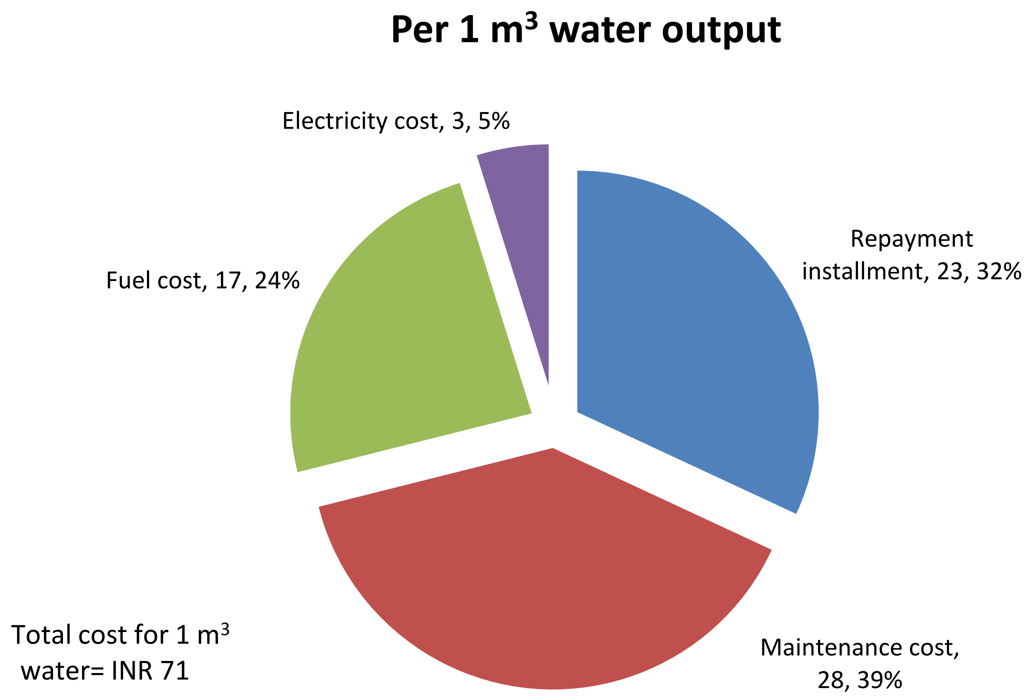


Fig. 10. Major cost components to produce 1 m^3 clean water (INR).

7. Conclusions and future work

The present study has shown possibilities to use thermal energy in desalination by direct coupling of the steam Rankine cycle to a batch-RO system. A spreadsheet model has been created using simple mechanical and thermal relationships. Using this model, a wide range of designs may be explored for different input conditions and output requirements from 2–3 m^3/ day up to hundreds of m^3/ day of water output. Any thermal source may be used to run the machine; thus it gives wide choices of fuel including biomass, solar energy or waste heat from industry. With salinity of feed water at 4000 ppm, gain output ratio ranges from 69–162 and recovery ratio ranges from 0.7–0.76, depending on the steam conditions and machine dimensions. Very low specific energy consumption, in the range of 4–6 kWh/m^3 based on thermal energy input, ensures low cost of treated water. A particular small-scale unit delivers 7 m^3 of water per day at the cost of INR 71/m^3, using steam pressure of about 11 bars generated from a biomass boiler. Use of locally available materials ensures low capital cost and ease of maintenance. In future, improved designs may be created to achieve more constant net driving pressure at the RO membrane. Use of the Organic Rankine Cycle by selection of appropriate working fluids may also be explored to run the power cylinder with a lower temperature heat source. It is planned to install a working model in India to carry out experiments with different input parametric conditions and thus validate the findings of this study.

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