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結合太陽能與海洋溫差於有機朗肯循環發電研究
研究成果報告(精簡版)

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結合太陽能與海洋溫差於有機朗肯循環發電研究

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Abstract

Rankine cycles using refrigerant- and benzene-series fluids as working fluid to convert power from renewable energy resources, such as solar energy and ocean thermal energy, are investigated presently. These fluids are R-11, R-12, R-113, R-114, R-123, R-152[a], R-500, R-502 and C_6H_6, C_7H_8, C_8H_{10}. First of all, the thermal-physical properties have been categorized. A language software "Mathcad" is employed in the calculations for the properties and thermodynamic behavior of the systems. Basically the turbine condition is kept at saturated vapor. Results indicate that overall the wet fluids with very steep saturated vapor curve in T-s diagram perform better conversion efficiency. Change of the condensation temperature does not perform significant influence amongst these fluids. Since the temperature range between the highest and the lowest operation conditions is not very large, the exit vapor would be very dry and the damage possibility to the blades of the turbine could be ignored. In the near shore locations, appropriate combination of these two resources is expected to have great profit in economics and environment.

Key Words: Organic Rankine cycle, Working fluids, Ocean thermal energy cycle, Solar energy

一、 前言

根據台電資料顯示，電力供應將一年一年吃緊，加上環保意識抬頭，新建電廠備受阻擾，在現有的條件下，如何開發更多電力，實在是重要。舉例來說，核能發電受到過度政治化的影響與扭曲，爭議性太大，因而已付出相當的社會成本，此外由於其發電廠規模較大，所需建廠時間較久，需長時間規劃，因此對於供電量的吃緊幫助有限。火力發電由於燃燒大量的石化燃料，因而排出大量的二氧化碳，不但造成空氣污染，地球溫室效應更是嚴重。所以開源的策略相當有困難，因此如何從環保的觀點下手，加強自然資源的利用，是目前較重要的問題。

事實上，太陽照射在地球上的熱源大都浪費掉，其使用也非常少見，若是觀察這些自然界中的溫度，其能量可轉換成動力的效率已經大可利用。以海洋溫差發電為例，台灣東部海域地形陡峻，離岸不遠處，水深即達 800 公尺，深層水溫約 5℃。同時海面有黑潮暖流通過，表層水溫達 25℃，使每年東部的海水溫差平均維持在 20℃以上，由於地形及海水條件俱佳，開發海水溫差發電的潛力雄厚。再以太陽能源為例，太陽照射至地球

的能量，僅是太陽能源 的廿二億分之一，這些廿二億分之一能源可供 全人類使用一萬七千五百倍的量，除了 架設的設施及運作成本外，太陽能可以說是全 球最乾淨、無污染且用不完的能源。況且台灣位於熱帶區，夏天陽光強烈，若以此為 ORC 的熱源，雖然效率可能較低，但這是一種無污染，不用熱源成本的好方法。相關有機朗肯循環(Organic Rankine Cycle, ORC)的研究，本研究團隊近年來也有相當的心得[1~3]。

ORC 於海洋溫差發電或太陽能發電系統是由蒸發器、渦輪機、發電機、凝結器、工作流體泵浦、熱交換器等組成，如圖 1 所示。選擇適合其溫度範圍的工作流體流入其蒸發器，並導入高溫熱源，使工作流體在熱交換器中因受高溫加熱而沸騰，其蒸汽經由連接管路送到渦輪機並使其轉動，並進而帶動連接的發電機發電。另方面，自渦輪機溢出的蒸汽則匯入凝結器，此時並導入低溫區，這些蒸汽因而冷卻成液態的工作流體而再次的由泵浦重新送至蒸發器。這樣的操作循環週而復始不斷進行。只要表、深層海水間存有溫差，太陽的照射能如此強烈，即能經由上述循環從海水中不斷獲得電力。

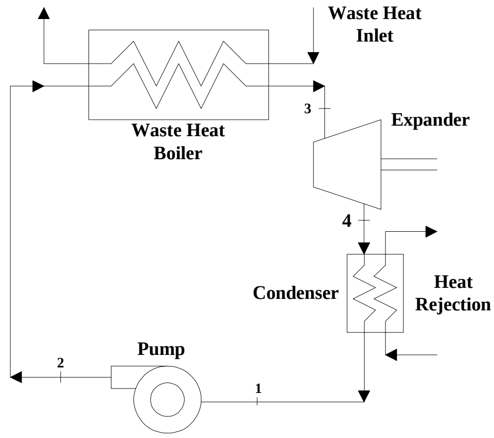

圖 1、有機朗肯循環示意圖

至於上述所提到的熱交換器，其材料的選擇是相當的重要，以海洋溫差發電為例，由[4]可知，材料遭受海洋的腐蝕相當嚴重，所以國內許多海洋建築物如跨海大橋、發電廠的冷凝管等均需考慮海洋的腐蝕。熱交換器的塑膠材質通常為 PPS、ABS、PA、PVC，而塑膠熱交換器的優點有：較高抗腐蝕性、抗化學性。重量輕、成本低、壽命長。但也有熱傳係數低、不耐高溫、強度較小等缺點。

二、 工作流體與選擇

由於有機工作流體的適用性較低，因此選擇工作流體之前需對各種工作流體的性質有深入的了解，來選擇最適當之工作流體。一些工作流體性質如表一所示，另外由[5-6]可知，在有機朗肯循環中，工作流體的熱物理性質和化學性質，不僅會影響系統效率的變化，同時對系統運轉時的穩定性與安全性，影響都相當地大。所以工作流體的選擇，是非常重要的工作。

表一、工作流體之基本熱力性質

Working fluid	R-11	R-12	R-152a	R-500	R-502
流體分類	濕流體	濕流體	濕流體	濕流體	濕流體
渦輪機出口狀態	液汽共存	液汽共存	液汽共存	液汽共存	液汽共存

臨界溫度（K）	471	111.8	389.4	378.6	355.2	
Critical pressure (MPa)	4.41	4.125	4.45	4.43	4.075	
Normal boiling temperature (K)	296.2	243.21	248.01	239.49	227.58	
h_{fg} @ 1 atm (kJ/kg)	178.8	166	318.43	200.81	172.48	
Working fluid	**R-113**	**R-114**	**R-123**	**C_6H_6**	**C_7H_8**	**C_8H_{10}**
流體分類	乾流體	乾流體	乾流體	乾流體	乾流體	乾流體
State at turbine exit	Super-heated	Super-heated	Super-heated	Saturated	Saturated	Saturated
Critical temperature (K)	487.3	419.03	456.9	562	591	616.2
Critical pressure (MPa)	3.41	3.261	3.67	4.89	4.119	3.51
Normal boiling temperature (K)	296.3	276.68	411.5	353	383.6	411
h_{fg} @ 1 atm (kJ/kg)	182	131.75	171.5	395.4	362.5	339.9

使用不同的工作流體，其適用範圍亦不相同，因此對不同的廢熱回收系統應選擇其適宜的工作流體，以增加系統之經濟效益。以下為對有機工作流體之選擇所需的考量：

1、Toxitisity of working fluid: 舉凡所有的有機流體皆具有毒性，因此為了防止不當的操作而造成洩漏，致使工作人員中毒，應選擇毒性較低的流體。

2、Chemical Stability: 有機流體在高溫高壓時會產生分解，對材料將會產生侵蝕，甚至極易引發爆炸或燃燒。所以系統在規劃時，必須將操作的工作溫度與壓力，設定在流體穩定而不至於發生危險的區域內，以防止產生破壞，或是縮短了流體與設備之使用壽命。

3、Boiling temperature of working fluid: 某些有機流體在常壓下的沸點相當的低，甚至低於水溫，若當作 ORC 系統的工作流體，冷凝器的冷卻物質的溫度必須隨著降低。

4、工作流體的燃點。較高的燃點，能降低燃燒的可能性，且可避免工作流體對整個系統的結構及元件導致危險的發生。

5、Specific heat: 較高的比熱值將會增加冷凝器的負荷，因此比熱值愈低愈好。

6、Latent heat: ORC 系統是利用飽和蒸汽或過熱蒸汽來作功，因此較高的潛熱值可以使熱回收效率提高。

7、Thermal conductivity: 較高的熱導性，有助於熱傳效果的增加。

本研究主要是以熱力性質的觀點來分析。根據以上所述，選出符合各種條件之工作流體，同時配合熱力性質而選出冷媒類流體為 R-11、R-12、R-113、R-114、R-123、R-152A、R-500、R-502 以及苯類流體為 C_6H_6、C_7H_8、C_8H_{10} 等十一種工作流體來加以研究。圖 2 中可以看出各類流體當溫度增加時，其熱物理性質的變化趨勢，而後將在結果與討論的部份基於此工作流體之物性，來判斷工作流體對系統效率的影響。

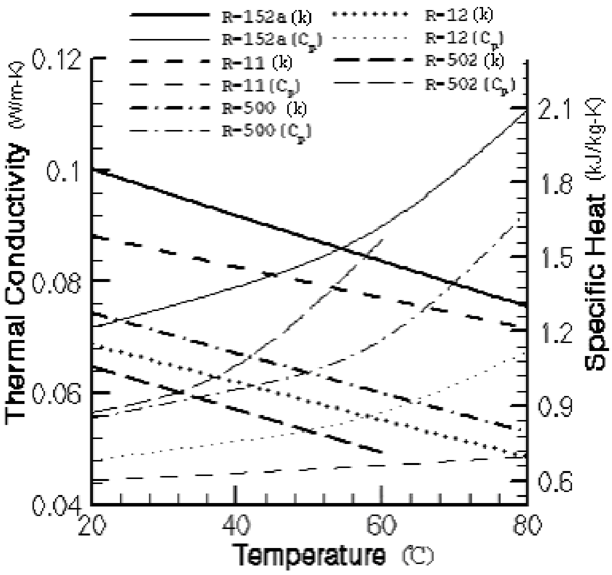

圖 2a、濕流體之物理性質（比熱與熱導性）

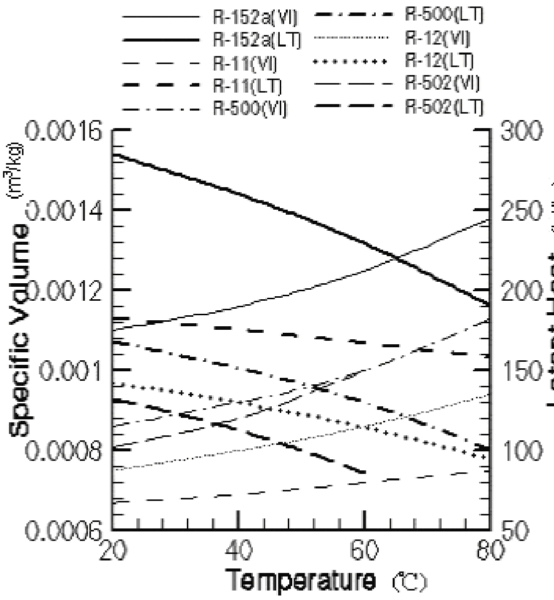

圖 2b、濕流體之物理性質（潛熱與比容）

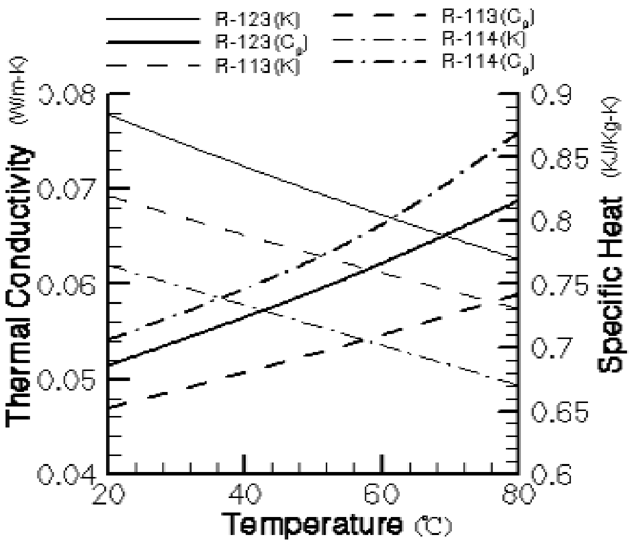

圖 2c、乾流體之物理性質(比熱與熱導性)

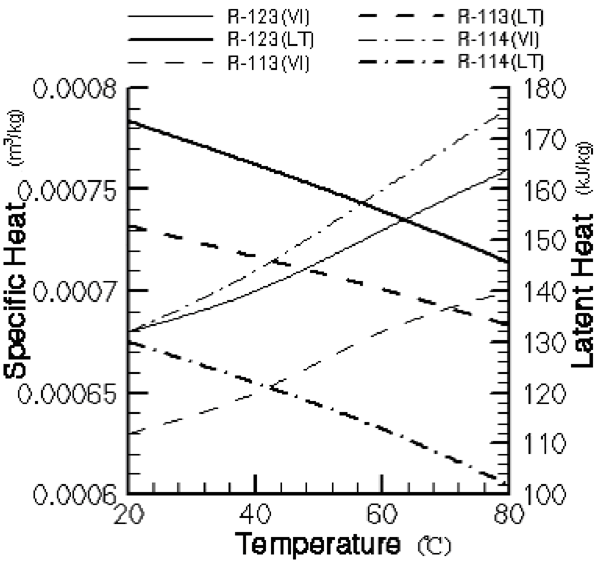

圖 2d、乾流體之物理性質(潛熱與比容)

圖 2、ORC 系統使用各類工作流體時，溫度與流體物理性質之間的關係

三、 數學分析模式：

以下的數學模式將探討應用 ORC 於再生能源的熱力分析。觀察其不同之工作流體對 T-s 圖的差異。如圖 3 所示，依據工作流體之飽和蒸汽曲線斜率為乾流體、等熵流體與濕流體，且此數學模式的設定分別為案例一為海水溫差及案例二為太陽能作熱力分析來探討。

另外要說明的是，在進行以下數學模式的分析與計算時，並不考慮蒸發器、冷凝器以及管路之中，因流體相變化以及流體與管壁間之磨擦所造成之壓力降等因素。圖 4 則是在

朗肯循環中，工作流體的各狀態位置以及渦輪機作功時的各種路徑。以下就理論分析的模式來進行分析。

泵浦：$w_{12}=(p_2-p_1)v_1$ （因狀態 2 在液態，比容變化不大 $v_1 \fallingdotseq v_2$）

$h_2=h_1+w_{12}$ ，$h_3=f(T_3,x_3)$

熱交換器： $q=(h_3-h_2)$

$h_4=f(p_4,s_4)$

蒸汽渦輪機：$w_{34}=(h_3-h_4)$

$\eta_t=1$ 整體效率：$\eta_{th}=\dfrac{w_{34}-w_{12}}{q_{23}}$

討論渦輪機效率：

$1 \sim x_{4a} \sim x_4$ $h_{4a}=f(p_3,x_{4a})$

$\eta_t=(h_3-h_4)/(h_3-h_{4a})$

$w_{34a}=(h_3-h_{4a})$

整體效率：$\eta_{tha}=\dfrac{w_{34a}-w_{12}}{q_{23}}$

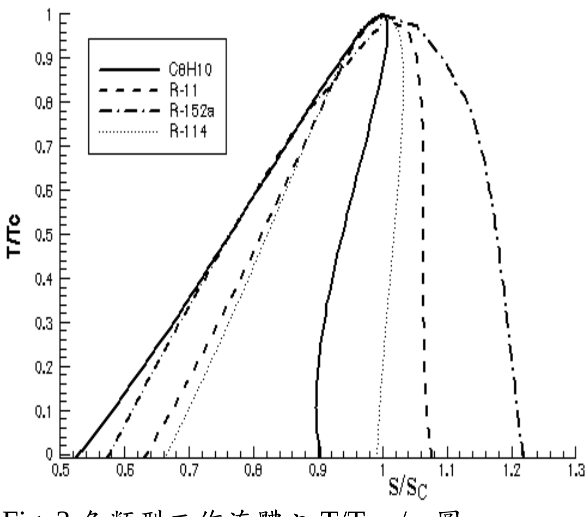

Fig. 3 各類型工作流體之 T/T_C-s/s_C 圖

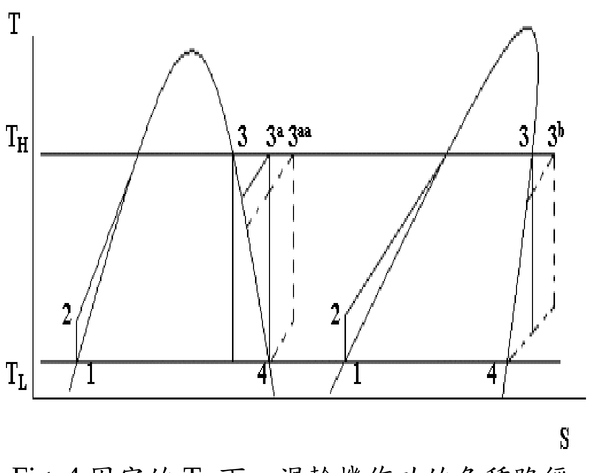

Fig. 4 固定的 T_H 下，渦輪機作功的各種路徑

四、 Results and Discussion

A computer program employing MATHCAD was developed to simulate the thermodynamic performance of the working fluids under various boundary conditions. The overall flow diagram is shown in Fig. 5. 在 ORC 系統裡，如圖 4 所示，當渦輪機入口處設定為狀態 3 時，即為飽和區或過熱區。由[7~8]得知，對一相同的工作流體而言，當熱源固定時，效率會隨著渦輪機進口壓力的增加而提升。明顯地，當狀態 3 在過熱區時對於系統效率而言，是不理想的；且過熱時的蒸發器所需的材質成本比在飽和區時貴得多；再者氣體的熱傳效果較差，加上熱量的損失，導致在過熱區所要達到預定的溫度，所需的熱源也要比飽和區來得高。

基於這些理由，故以下之研究方向將渦輪機入口處設定為飽和蒸汽狀態，同時選擇上述之十一種流體來搭配 ORC 系統。本研究將分成海洋溫差與太陽能熱源為兩案例來進行熱力分析。

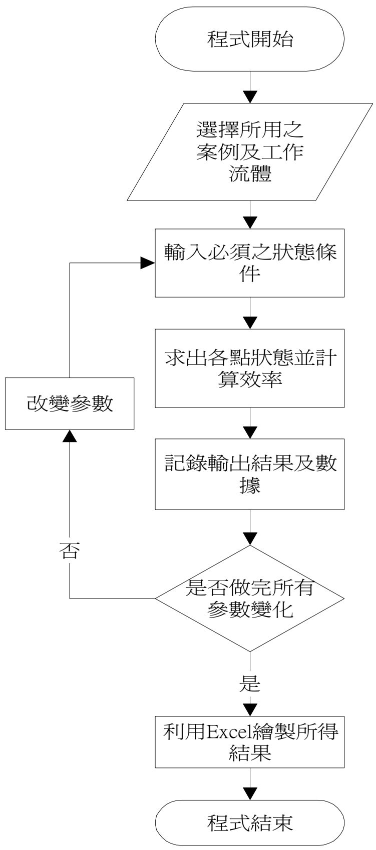

Fig. 5 計算 ORC 系統效率之 Mathcad 流程

4.1 案例一：OTEC

此案例是利用表、深層海水間存在的溫度差來供給 ORC 系統熱源。根據上述之分析，在工作流體的選擇方面，可依據工作流體的飽和曲線特性與不同的渦輪機入口溫度，其流體的效率趨勢產生的交點來作為研究的依據。其操作條件為，升壓泵入口溫度

設定為 5℃，渦輪機入口溫度設定在 20~40℃作變化。從圖 6 與圖 7 中可以發現，當渦輪機入口溫度提高時，系統效率會隨之上升。這是因為當渦輪機入口溫度提高，工作流體從蒸發器中吸收的熱能大幅提升。圖 6 則是說明當渦輪機入口溫度改變時，對不同乾流體的效率趨勢。明顯地可以發現，在渦輪機入口溫度範圍在 20~23℃時，苯類與冷媒類之工作流體在效率上並無太大的差異；當溫度超過 23℃時，苯類工作流體在效率上會有較佳的表現。若從工作流體之飽和蒸汽曲線走向可以發現，苯類流體雖然屬於乾流體，然而在此較低的操作溫度條件下，飽和蒸汽曲線已呈現近似等熵的濕流體。如圖 3 中所示，可以很明顯地發現 C_8H_{10} 在較低溫時，飽和蒸汽曲線的走向由正斜率漸漸變成負斜率。另外可以發現到，R-113 與 R-123 在溫度約為 30℃時，會有一交點產生。當渦輪機出口溫度範圍在 20~30℃時，R-123 在系統效率上的表現較優於 R-113；相反地，當溫度範圍在 30~40℃時，R-113 的系統效率較高。

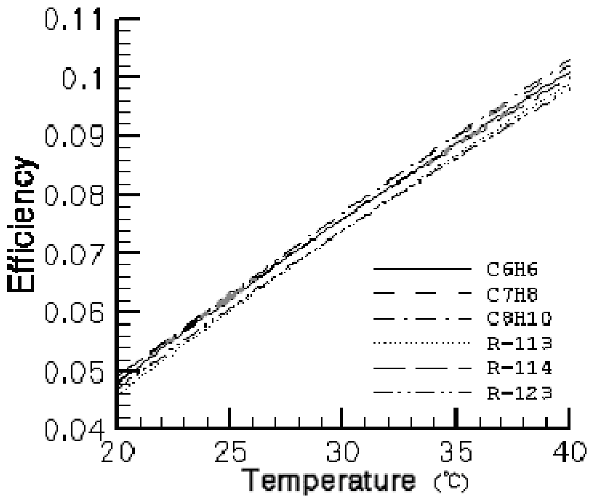

圖 6、案例一：工作流體為乾流體時，系統效率和渦輪機入口溫度之間的關係

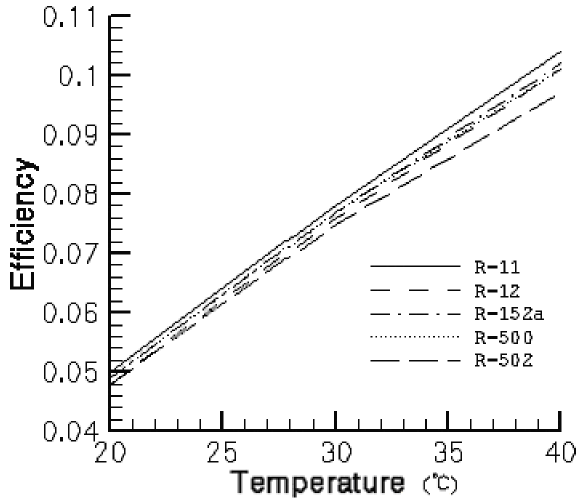

圖 7、案例一：工作流體為濕流體時，系統效率和渦輪機入口溫度之間的關係

針對流體的物性可以來判斷，如圖 2 所示。R-123 的熱導性和潛熱值皆高於 R-113，亦即在循環系統中，較高的熱導性和潛熱值其熱傳效果與熱回收率皆會相對的提升。然而，從比熱值的趨勢來看，R-123 較 R-113 來得高，且當溫度增加時，增加的趨勢雖不明顯，但增加的速度較快。以熱力學的觀點來分析，當乾流體在渦輪機出口處做等熵膨脹過程，冷凝器入口為過熱狀態。在此說明，本研究分析並不考量使用再生器（regenerator）。因此，工作流體之比熱值愈高，對於冷凝器的負荷會隨之增加，系統效率則會降低。同時也必須考慮工作流體在低壓區的潛熱寬度大小，寬度愈寬，亦即有更多的能量經由冷凝器排掉。由於此案例是在較低的溫度範圍下操作，低壓區的潛熱寬度影響較小，然而隨著渦輪機入口溫度的增加，潛熱寬度的影響對整體效率的趨勢會逐漸的明顯。因此，以工作流體之各項物性條件來判斷，會產生此一交點。而利用此交點在不同熱源下選擇適當的工作流體，是一重要的依據。

苯類流體效率有較好的趨勢，是因為冷媒類流體通常具有較低的沸點，即在相同的冷凝溫度下，冷媒類流體的蒸汽壓力較高。參考圖 4 可得知，當狀態一的壓力較高時，朗肯循環的四個狀態所圍成的有效面積較小，以熱力學的觀點來說，則會造成系統效率的下降。圖 6 中不同的乾流體所呈現的效率趨勢，可以依據苯類流體在較低溫的操作條件下之飽和曲線的特性與不同的海水溫差來選擇適當的工作流體。

圖 7 則是說明當渦輪機入口溫度改變時，對不同濕流體的效率趨勢。從各工作流體之飽和曲線觀察可以發現，R-11 與 R-12 是近似等熵的濕流體，亦即飽和蒸汽曲線斜率趨近於無窮大。在實際運轉測試時，倘若能控制得宜，渦輪機出口處工作流體的狀態會落在飽和曲線上。此種流體有兩種好處，一是不會在等熵膨脹過程中發生液滴的現象；另一是不需加裝再生器來減少冷凝器的負擔。在經過熱力分析後，明顯地可以發現此種工作流體的效率較佳。

由於工作流體的性質對 ORC 系統影響甚大，若單純只考慮等熵流體的效率較佳為基準並不客觀。以 R-12 為例，便是一例證。如圖 2 所示，R-12 的熱導性和潛熱值相對於其他工作流體要來的低，對於 ORC 系統而言，熱傳效果與熱回收率較低的影響，使得整體效率趨勢的表現並不佳。綜觀各項物性因素考量，R-502 的效率為最低。而在此可以發現到渦輪機出口溫度在約 33℃時，R-152a 與 R-500 的工作流體會出現一交點。當溫度範圍在 20~33℃時，此兩種流體的效率趨勢近乎相同；然而當溫度超過 33℃時，R-500 的效率很明顯地會降低。以工作流體之各項物化性質與工作流體之飽和曲線來考量，隨渦輪機入口溫度增加的潛熱值，直接地影響循環系統的熱回收率，對於系統效率而言，R-152a 的效率趨勢較佳。

圖 8 說明當渦輪機入口溫度為一定值時，改變冷凝器的溫度範圍與系統效率的關係。由 T-s 圖來看，當冷凝器出口溫度提升，其代表系統所排出的熱量勢必要增加，因此使得系統效率下降。圖中很明顯地可以發現，系統效率隨著冷凝溫度的下降而上升。然而，不同的工作流體所呈現的效率趨勢，並沒有太大的差異，趨勢的走向幾乎相同。

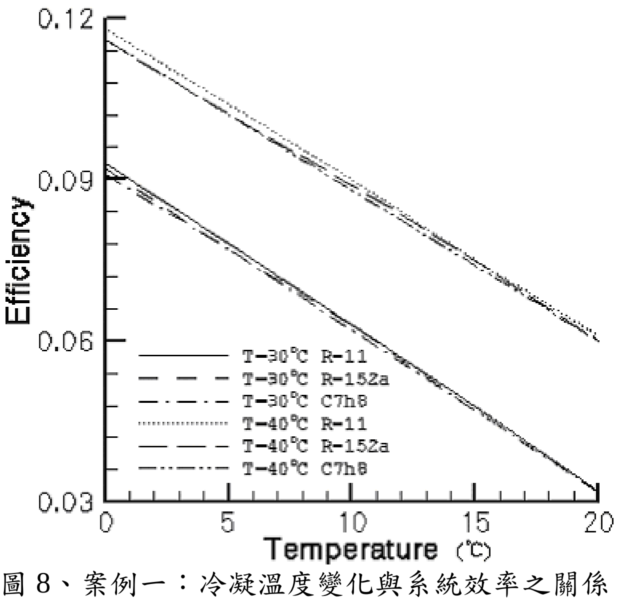

圖 8、案例一：冷凝溫度變化與系統效率之關係

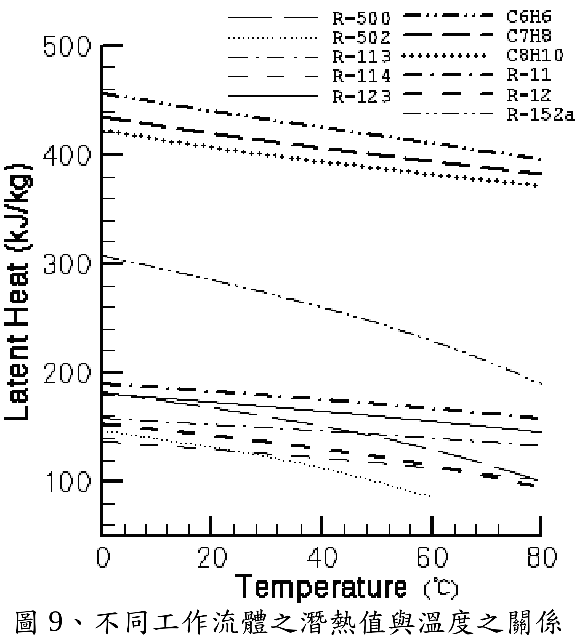

圖 9、不同工作流體之潛熱值與溫度之關係

將此設定的操作條件與上述固定冷凝溫度之操作條件的趨勢，作一比對。可以發現，改變冷凝溫度範圍對於工作流體方面的選擇，變得更加的廣泛。當改變渦輪機入口溫度時，在流體選擇上考量點相當的多；相對地，改變冷凝溫度範圍對於流體的選擇，只需考量工作流體的穩定性，安全性及對於環境的影響等相關的化學性質。對於設計循環系統而言，冷凝溫度的改變也是一重要的參考因素。

依據上述的分析，從工作流體之物性以及基本的熱力性質，可以歸類三項於工作流

體在不同溫度時，對系統效率影響的主要因素：飽和曲線的走向、比熱與潛熱。其中可以很明顯地發現，工作流體在不同溫度時的潛熱值影響較大。因為工作流體的潛熱值對效率而言，不僅影響蒸發器所吸收的熱能；同時在低溫時的潛熱寬度，對冷凝器所需排除的熱量也會影響系統的效率。圖 9 是將各工作流體的潛熱值在不同溫度範圍下所呈現的趨勢。從圖中可以發現，系統效率表現較佳的流體其潛熱值都較高，且隨著溫度的增加，其趨勢則較為平緩。雖然 R-152a 在效率上比其他的濕流體要來得高，不過當所設定的溫度增加時，其潛熱值下降得比其他流體快，系統效率則降低。然而對於本研究之操作條件較低，R-152a 的潛熱值隨著溫度的增加，對於系統效率而言，其實影響並不明顯。因此，可以根據圖 9 所得知，工作流體的選擇，潛熱值的大小可以為一重要的參數來考量。然而，潛熱值的大小及寬度對於系統效率而言，影響較其他物性及熱力性質來得大，但也必須針對其他的物理性質與潛熱值互相作一比對，就實質上來說，是比較客觀的。

由上述的結果可以得知，近似等熵流體的效率表現較佳，濕流體在系統效率方面也有不錯的表現。然而對濕流體而言，在實際操作上倘若無法循著飽和蒸汽曲線前進，工作流體處於液汽共存的狀態下，對渦輪機葉片是一種極大的傷害。因此，以下將針對濕流體的乾度（quality）及渦輪機效率之間來做探討，如圖 10 所示。

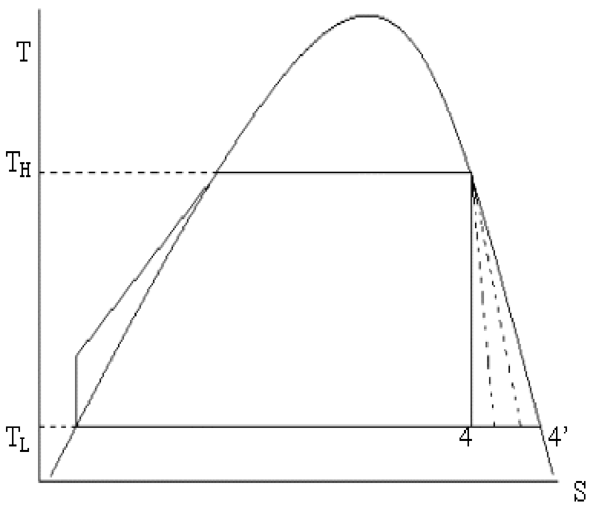

圖 10、濕流體之渦輪機效率變化區間

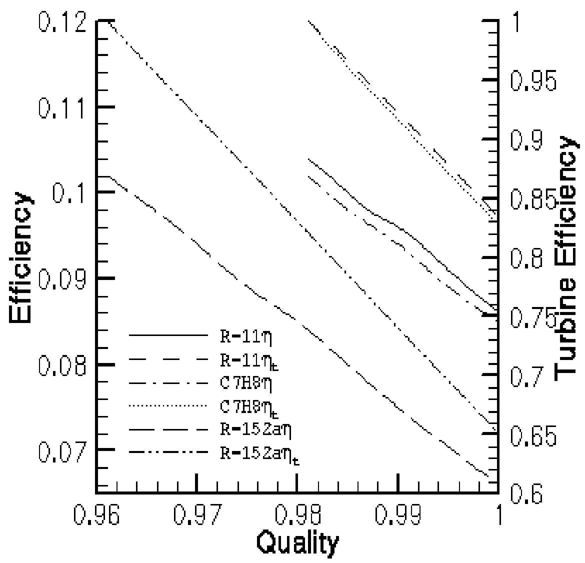

圖 11、案例一：乾度與渦輪機效率之間的影響

此案例所設定的溫度操作條件為：低壓區為 5℃，高壓區為 40℃。且當工作流體在渦輪機入口時，處於飽和蒸汽的狀態。工作流體選擇方面，則依據上述所分析，選擇效率表現較佳的流體。圖 11 說明此案例的操作條件下對於乾度、系統效率與渦輪機效率之間的影響。從圖中可以發現，當乾度為 1 時，亦即在設計渦輪機效率時，效率的要求不須太高，R-152a 流體的系統效率約為 6.7% ，R-11 流體的系統效率約為 8.6% ，甲苯的系統效率約為 8.5% 。若以渦輪機效率相對於系統效率而言，R-11 流體在設計渦輪機效率上較甲苯的要求來得低。從飽和蒸汽曲線的走向可以看出一趨勢，當飽和蒸汽曲線斜率趨近於無窮大，亦即工作流體為近似等熵流體，工作流體經由等熵膨脹過程至低壓區，所產生的乾度近似於 1，實際運轉只要能適宜地操作，再加上此類流體的特性，因此在渦輪機設計上的要求度不高。然而，當工作流體經由等熵膨脹過程至低壓區的乾度愈小，渦輪

機的效率則必須愈低才能達到乾度為 1，對於系統效率而言則相對地降低。由於 R-152a 飽和蒸汽線斜率相當平緩，導致渦輪機出口處乾度太小，如果要改善此缺失，也可將渦輪機入口狀態操作於過熱區，如圖 4（3～3^a）。實際上也可因由渦輪機效率小於 100%而將之出口處設計在飽和蒸汽線上，讓 R-152a 可以在低溫範圍使用。

4.2 案例二：Solar energy as high-temperature reservoir

此案例是利用太陽能產生的熱量來供給 ORC 系統熱源。其操作條件為，升壓泵入口溫度設定為 20℃，渦輪機入口溫度設定在 40~60℃作變化。而分析的結果得知苯類工作流體的效率較冷媒類要來得好；且濕流體仍 R-11 較好，此趨勢和案例一的結果雷同。圖 12 說明此案例的操作條件下對於乾度、系統效率與渦輪機效率之間的影響。將此案例的整體趨勢與案例一作比較，可以發現此兩案例趨勢略微不同，差別就在於流體在不同的溫度範圍間（T_H 到 T_L）有其不同之飽和蒸汽線斜率，以致於渦輪機出口產生不同的乾度。因此，在考量系統效率與如何設計符合所需求的渦輪機效率，可以依據圖 12 的效率趨勢，以及與工作流體間的選擇作一整合，相信對於再生能源與 ORC 的應用上更能發揮得更廣泛。

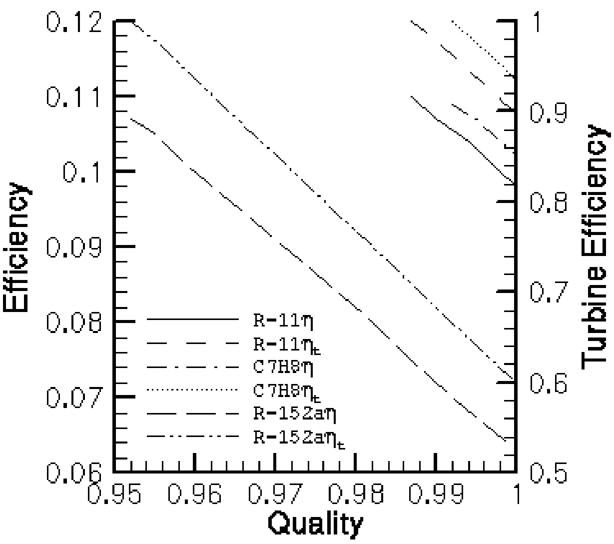

圖 12、案例二：乾度對渦輪機效率之間的影響

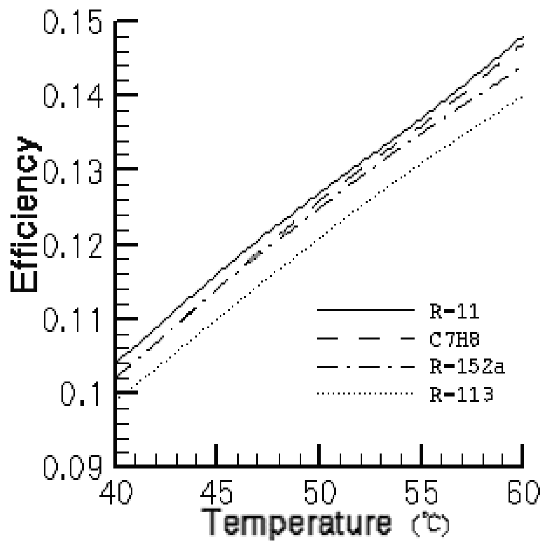

圖 13、兩案例搭配時，系統效率和渦輪機入口溫度之間的關係

從此兩案例可以發現，工作流體為乾流體時，苯類流體的效率趨勢較佳。雖苯類流體屬於乾流體性質，然而在此兩案例的操作條件下，其飽和蒸汽曲線卻是近似等熵的濕流體。且工作流體經等熵膨脹後，工作流體在渦輪機出口處之乾度極近似於 1。對於案例一而言，苯類流體效率為 5% ~10% ；對於案例二而言，苯類流體效率為 6.2% ~11% 。當工作流體為濕流體時，除了飽和蒸汽曲線斜率為影響效率的主要因素外，渦輪機入口溫度增加時的潛熱值對系統效率而言，也是必需要考量的重點。當改變冷凝溫度時，流體對系統效率而言，並未有明顯的差異。由於太陽能熱源是利用太陽照射類似太陽能熱水器的表面；海洋溫差的散熱是以低溫冷海水，因此可以考慮將此兩案例作一結合，亦即高、低溫區之間的有效面積加大。根據所分析的效率作加成，若取高溫於 40～60℃，低溫於 5℃，而整體的效率將可達到 11%~15% 。在此分析而言，如圖 13，近似等熵的濕流

體能有較佳的效率。另外由此圖可看出 R-11 的效率高於 R-113，若以 T-s 圖來看，R-113 為乾流體，在本分析時將使渦輪機操作出口落點處於過熱蒸汽，造成 T-s 圖上的缺角而使效率不佳。而以物性來看，R-11 之熱導性和潛熱值也高於 R-113。因此，選擇適當的流體搭配此兩案例做結合。在經濟效益上不僅可以節省成本，同時可達到高效率的運作。

五、 Conclusion

目前完成了所選擇之十一種有機工作流體的分析模式，分別應用於不同熱源，海洋溫差與太陽能熱源。經由熱力分析與各類有機流體之物性來判斷。在本研究兩案例中，適當的工作流體選擇與各流體的飽和曲線特性，以及在不同熱源範圍下，從效率圖上可以發現每個流體在其適當的熱源下，可以得到其有效的使用範圍。同時也可得知在實際運轉時，針對系統的設備來進行改善，對系統之整體效率而言，是一大幫助。乾流體於本研究的缺點在於會使渦輪機操作出口落點處於過熱蒸汽，造成 T-s 圖上的缺角而使效率沒有提升。何況過熱蒸汽將使冷凝器太過於負擔而需使用再生器來改善，但設備成本也因此較高；濕流體的缺點就在於液滴的產生，將破壞渦輪機；近似等熵的濕流體其較無渦輪機破壞之顧慮，也無須像乾流體多裝再生器來減少冷凝器的負擔。因此若以近似等熵的濕流體作為本研究的工作流體，其考慮的重點僅在於流體的成本、穩定性、安全性等等，是 ORC 應用於自然界溫度最佳的工作流體。 同時對於此兩案例的結合，未來將進一步地作熱力分析，以求其有高效率的表現。

六、Nomenclature

h	enthalpy	C_p	Specific heat
h_{4a}	turbine exit enthalpy for irreversible process	h_{fg}	Latent heat
k	thermal conductivity	v_l	liquid specific volume
p	pressure	η_{th}	Thermal efficiency
s	entropy		
T	temperature	subscript	
v	Specific volume	1~4	Location of the state
x	dryness	t	turbine
w_{12}	work done by the pump		
w_{34}	Work done by the turbine		
w_{34a}	Work done by the turbine for irreversible process		
q_{23}	heat added to the evaporator		

七、參考文獻

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會議心得報告：

※ 會議規模(請勾選)：X□全球性 □ 區域性。
與會國家／人數：約__20______國／___500______人。

一、研習心得
二、會議（任務）執行情形
三、返（回）國後需要處理之工作要項
四、建議之工作計畫

一、 研習心得：

本次會議不但瞭解到世界各國對海洋能應用之發展現況，並且發現國內數所大學亦做相類似之研究，故可就近互相觀摩，吸引他人經驗，以改進未來之研究.

二、 會議執行情形：

本次會議為泰國 Asian Institute of Technology 及 Provincial Electricity Authority 主辦，美國 IEEE 協辦之國際會議，並為國科會鼓勵國科會計畫之學者參加。會議之主題為能源之策略與規劃，尤其是再生能源，但其包括範圍相當廣泛，因泰國海岸線長，故海洋能應用為該國際會議之重點之一. 國內參加的有本校，建國科大，台東大學，清華大學等校，在海洋能應用上應可互相觀摩. 本人之報告主要是利用海水高低溫差推動有機朗肯循環發電，在理論上已證實有 ~12-15% 之發電效率，尤其正在研究之太陽能池，將能更提昇發電效率，且引起其他學校之興趣.

三、 反國後需要處理之工作要項：

由此大會中，大部分學者對於海洋能應用均充滿希望，但共同認知為此類研究計畫應由政府主導，而各研究單位應相互交流，以發揮最大之研發動能，避免研究資源浪費. 反國後需要處理之工作要項有:

(a) 收集國內外最近之相關研究報告

(b) 連繫國內此研究之專家學者，交流研發經驗

(c) 持續改善本人之海洋能發電效率，並進行其參數最佳化分析

(d) 繼續太陽能熱水池之研發，以解決海洋表面溫度不夠高之缺點，提升發電效率

四、 建議之工作計畫：

今後之工作項目，首要乃繼續太陽能熱水池之研究，以取代海洋表面溫海水，以提高海洋能發電效率. 並計畫邀請建國科大，台東大學在此方面之專家學者到系上交流，分享心得.

Renewable Energy from the Sea - Organic Rankine Cycle using Ocean Thermal Energy Conversion

S. K. WANG[1]*, T. C. HUNG[1]

Abstract *–Rankine cycles using refrigerant- and benzene-series fluids as working fluids in converting low-grade energy from renewable energy resources such as solar energy and ocean thermal energy were investigated in this study. The main purpose is to verify the feasibility of utilizing ocean energy (i.e., ocean thermal energy conversion, OTEC) which can also be combined with solar energy in an organic Rankine Cycle (ORC) to generate electricity. Parameters under investigation were turbine inlet temperature, turbine inlet pressure, condenser exit temperature, turbine exit quality, overall irrversibility, and system efficiency. Results indicate that wet fluids with very steep saturated vapor curves in T-s diagram have a better overall performance in energy conversion efficiencies than that of dry fluids. It can also be shown that all the working fluids have a similar behavior of the efficiency-condenser exit temperature relationship. Furthermore, an appropriate combination of solar energy and an ORC system with a higher turbine inlet temperature and a lower condenser temperature (as operated deeply under sea level) would provide an economically feasible and environment-friendly renewable energy conversion system.*

Keywords – **Ocean thermal energy conversation, Organic Rankine cycle, Solar energy, Working fluids,**.

1. INTRODUCTION

Low-grade heat from renewable energy sources is considered to be a good candidate to generate electricity. Among those sources, OTEC and solar energy are typically utilized in converting low-grade heat into power generation and other applications. OTEC systems use the ocean's natural thermal gradient to drive a power-producing cycle. As long as the temperature difference between the warm surface water and the cold deep water is greater than about 20°C (36°F), an OTEC system can produce a significant amount of power. Lennard [1] identified the best locations for OTEC around the world, and he found that the natural ocean thermal gradients necessary for OTEC operation generally exist between latitudes 20 deg N and 20 deg S. On the east coast of Taiwan, a steep offshore slope provides a good environment for the application of OTEC as a water depth of 800 meters and a water temperature of 5℃ can easily be found near shore. Tanner [2] did an extensive review and concluded that Taiwan is one of the best markets for utilizing OTEC, and he also suggested some potential applications of OTEC such as fresh water production in desalination. Tseng et al. [3] used a numerical Sequential Quadratic Programming scheme to obtain the optimum power output of an OTEC plant.

Takazawa et al. [4] performed experiments on a barometric-type open-cycle OTEC system, and they concluded that the existence of non-condensable gas may reduce the efficiency of the direct contact heat exchanger when the concentration of the non- condensable gas is above 8%. In spite of a pessimistic conclusion made by Odum [5] who used an emergy analysis (emergy ratio indicates a net contribution of electric power) to suggest that OTEC is not likely to become economical in Taiwan, OTEC is still considered to be a potentially viable option in power generation. Using ammonia as the working fluid, Yeh et al. [6] theoretically investigated the effects of the temperature and flow rate of cold seawater on the net output of an OTEC plant. They concluded that a maximum output of the net work exists at a certain seawater flow rate. Using R-12 as the working fluid, Wu and Burke [7] used specific power – power per unit total heat exchanger surface area – of a heat engine to be the objective function in the design of an OTEC Rankine power plant. Through manipulation of boiler pressure and condenser pressure, the specific power of the OTEC plant was calculated and an upper bound was determined. Madhawa Hettiarachchi et al. [8] proposed a cost-effective optimum design criterion for ORCs. Regardless of using geothermal heat source instead of OTEC or solar energy, they found that the choice of working fluid can greatly affect the objective function which is a measure of power plant cost – in some instances the differences could be more than twice.

Solar energy on the other hand has long been recognized as a tangible and viable source of energy. Despite its low temperature in energy collecting unit and low energy conversion efficiency, solar energy suits perfectly to act as the heat source in the evaporator of some low-grade heat recovery system such as ORCs. However, combined cycles using solar energy and OTEC in power generation have not received too much attention until recently. Straatman et al. [9] proposed a conceptual design of a hybrid OTEC-offshore solar pond power plant as compared with the pure solar thermal

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electricity plant, and they claimed the hybrid plant can reach an improved efficiency of 12%. Yamada et al. [10] performed a computer simulation of a solar-boosted OTEC system (SOTEC). The results show that the proposed SOTEC plant can enhance the annual mean net thermal efficiency up to a value that is 1.5 times higher than that of the conventional OTEC plant.

ORCs have been investigated for power generation for years. Several ORC systems have been installed for recovering waste heat and widely used for converting renewable energy into power. Hung et al. [11] used some cryogens as working fluids in an ORC operated between two isobaric curves, and they found that the system efficiency increases and decreases for wet and dry fluids, respectively, when turbine inlet temperature – the main parameter under consideration – increases. Liu et al. [12] investigated the effects of several working fluids on an ORC for waste heat recovery. They found that the presence of hydrogen bond in certain molecules such as ammonia, water, and ethanol may cause these fluids behave like wet fluids due to their large vaporization enthalpies, and these fluids are regarded as inappropriate for ORC systems. As stated in the previously mentioned studies, ORCs using OTEC as the high and low temperature ends are potentially feasible in recovering low-grade energy and generating power if adequate working fluids are used. The efficiency can further be improved if the high temperature end of the cycle is boosted by solar energy. The objective of this study is to gain a comprehensive understanding of the thermodynamic performances of an ORC using various working fluids. System efficiencies are calculated for an ORC using OTEC as the heat source and sink with and without the boost of solar energy. The following analyses focus on thermodynamic performances of the ORCs as scoping calculations without considering detailed system integration, e.g., the solar thermal pond or solar energy collector served as a boundary condition of the inlet temperature of the evaporator. Detailed calculations of pressure losses and heat transfer in evaporator and condenser are also ignored since they depend strongly on materials and configurations of the system components. Instead, irreveribilities of the working fluids in various major components of the cycle are calculated to evaluate the effects of those losses. Parameters under consideration are turbine inlet temperature, turbine inlet pressure, condenser exit temperature, turbine exit quality, overall irrversibility, and system efficiency.

An ORC system using low-grade energy sources is depicted in Fig. 1. The system is composed of an evaporator (waste heat boiler), a turbine expander, a condenser, and a pump. A working fluid flows into the evaporator in which the high-temperature heat source (which may come from the warm seawater or a solar pond) is utilized. The vapor of the boiling fluid enters the turbine expander and generates power. The exit fluid from the turbine expander then enters the condenser in which the low-temperature cooling water (i.e., the cold seawater) is utilized to condense the fluid. Finally, a fluid pump raises fluid pressure and feeds the fluid into the evaporator to complete the cycle. So long as a temperature difference between the high- and low-temperature ends is large enough, the cycle will continue to operate and generate power. The objective of this study is focused on thermodynamic analyses of the working fluids and the overall system efficiency rather than hardware arrangements such as the system integration of solar energy and OTEC. Therefore, issues regarding material selections, component configurations, frictional losses, heat transfer performances of the evaporator and condenser, and cost analysis are not considered in this study.

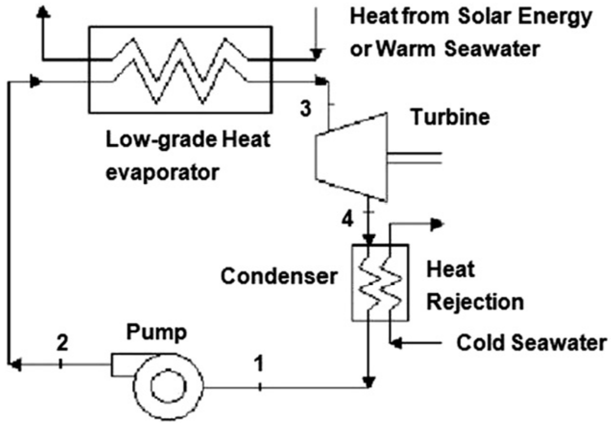

Fig. 1 A schematic Flow Diagram of an ORC system

2. Selection of working fluids

In an ORC, a suitable selection of the working fluids is a critical factor for achieving an efficient and a safe operation. Each working fluid has its own range of applicability according to its thermophysical properties under the considerations of a high efficiency and a safe operation. Important factors of the working fluids needed to be considered are listed below:

1. Toxicity of working fluid: All organic fluids are inevitably toxic. A working fluid with a low toxicity should be used to protect the personnel from the threat of contamination in case of a fluid leakage.
2. Chemical stability: Under a high pressure and temperature, organic fluids tend to decompose, resulting in material corrosion and possible detonation and ignition. Therefore a chemically-stable working fluid operated under working conditions should be selected.
3. Boiling temperature: Some of the organic fluids have a very low boiling temperature under atmospheric pressure. For those fluids, the temperature of cooling water in the condenser should be reduced. This can result in a more stringent requirement for the selection of the condenser.
4. Flash point: A working fluid with a high flash point should be used in order to avoid flammability.
5. Specific heat: A high value of specific heat represents a high load for the condenser. Hence a working fluid with a low specific heat should be used.
6. Latent heat: A working fluid with a high latent heat should be used in order to raise the efficiency of heat recovery.
7. Thermal conductivity: A high conductivity represents a better heat transfer in heat-exchange components.

The fluids under consideration in this study are refrigerant-series fluids such as R–11, R–12, R–113, R–114, R–123, R–152a, R–500, and R–502; and benzene-series fluids such as C6H6, C7H8, and C8H10. The functional dependences of temperature will be used in analyzing the system efficiency in this study.

3. Mathematical Analysis

The following mathematical model is used to analyze thermodynamic behavior of ORC systems. The slopes of saturated vapor curves in the T-s diagrams are used to identify the types of the working fluids (i.e., wet fluids and dry fluids) as shown in Fig. 2. Pressure drops occurred in various components and pipes are not considered in this model. Figure 3 shows the conditions of working fluids at various locations and paths of power generation in an ORC.

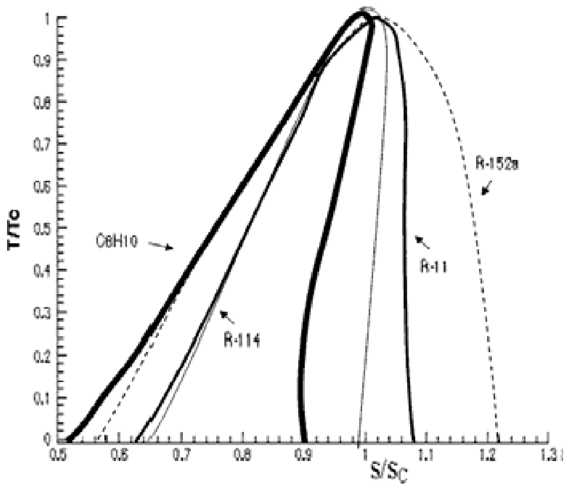

Fig. 2 T/T_c-s/s_c Diagram of Working Fluids

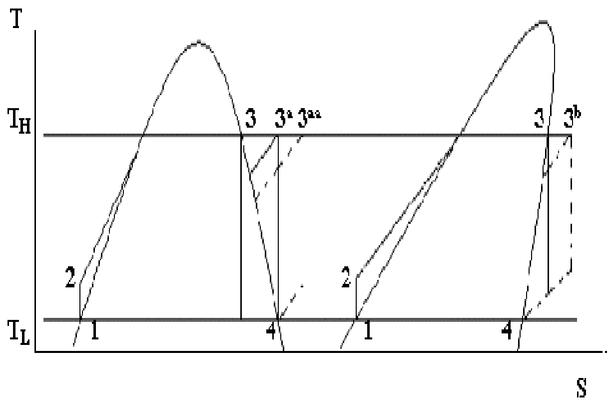

Fig. 3 T-s Diagram of Working Fluids in Turbine under a Fixed T_H

The mathematical model is analyzed as

Pump:

$W_{12} = (p_1 – p_2)v_1$

(1)

$H_2 = h_1 + w_{12}$ $h = f (T_3, x_3)$

(2)

Heat exchanger:

$Q_{23} = h_3 – h_2$ $h_4 = f (p_4, s_4)$

(3)

Turbine expander:

$W_{34} = h_3 – h_4$

(4)

Overall efficiency： $\eta_{th} = (w_{34} – w_{12})/q_{23}$

(5)

Turbine efficiency: $x_{4a} \sim x_4 \sim 1$

$h_{4a} = f (p_3, x_{4a})$

(6)

$\eta_t = (h_3 – h_4)/(h_3 – h_{4a})$

(7)

$w_{34a} = h_3 – h_{4a}$

(8)

Overall efficiency：$\eta = (w_{34a} – w_{12})/q_{23}$

(9)

Practically, due to irreversibility in an actual thermodynamic system, it is impossible to convert all the available thermal energy into useful work. Furthermore, irreversibility provides an additional means of estimating the system efficiency of a thermodynamic cycle. From the second law of thermodynamics, the equation of irreversibility rate can be expressed for uniform flow as follows:

$$\dot{I} = T_o \frac{ds_{tot}}{dt} = \dot{m}T_o[\sum_{exit} s - \sum_{inlet} s + \frac{ds_{sys}}{dt} + \sum_j \frac{q_j}{T_j}]$$

(10)

Assuming that the system reaches a steady state, and there is only one exit and one inlet for any component, Equation (10) becomes

$$\dot{I} = \dot{m}T_o[(s_{exit} - s_{inlet}) + \frac{q}{T}]$$

(11)

where T_o is the ambient temperature. Since the major contributions of the irreversibilities are from the processes 1-2 and 3-4, the total irreversibility rate becomes

$$\dot{I}_{tot} = \sum_j \dot{I}_j \cong \dot{m}T_o[-\frac{h_3 - h_4}{T_H} + \frac{h_{4a} - h_1}{T_L}]$$

(12)

From Eq. (12), one can see that the heat transfer rates in the evaporator and condenser associated with the ambient temperature are the key factors affecting the overall irreversibility; and accordingly, the system efficiency. For the sake of a better understanding of the effects of pressure on irreversibility, the availability ratio, φ, is defined as follows

$$\phi = \frac{\dot{m}q_{23} - \dot{I}_{tot}}{\dot{m}q_{23}}$$

(13)

The factor is the ratio of the available energy to the total energy obtained from the heat source.

4. Results and discussion

A computer program employing MATHCAD was developed to simulate the thermodynamic performances of the working fluids under various working conditions. As shown in Fig. 3, the turbine inlet condition is assumed to fall on State 3 which will be in saturated or superheated region. Material requirements of the evaporator are more stringent as the working fluid becomes superheated, and a lower thermal conductivity of the superheated vapor would result in a lower heat transfer rate as compared with the saturated vapor. Also, a heat source with a higher temperature for the evaporator is required if the fluid entering the turbine is superheated. Therefore, the working fluid is assumed to be saturated with its corresponding saturation pressure at the inlet of the turbine. The following analyses will be based on two types of energy resources: OTEC and solar energy.

4.1. Case 1: using OTEC as energy source

An ORC system using OTEC operates between a high temperature (supplied by the warm seawater) in the evaporator and a low temperature (supplied by the cold seawater in deep sea) in the condenser. Based on the slopes of saturated vapor curves and the inlet temperatures of the turbine, system efficiencies using various working fluids can be estimated. The inlet temperature of the pump is fixed at 5 ℃, i.e., heat transfer in the condenser is assumed to be very efficient, and the inlet temperature of the turbine is varied from 20 to 40℃ to simulate the heat source in the calculations. Due to a more amount of energy is received in the evaporator, system efficiency increases nearly linearly as the turbine inlet temperature increases for every working fluid under investigation as shown in Figs. 4 and 5. For dry fluids as shown in Fig. 5, refrigerant- and benzene-series have almost the same efficiencies when the turbine inlet temperature is low. As the turbine inlet temperature increases, benzene-series in general have a better performance in system efficiency since their saturated vapor curves become almost identical to those of the isentropic wet fluids. This can be seen in Fig. 3; C_8H_{10}, for example, has a saturated vapor curve with a positive slope changing to a negative slope as temperature decreases. Also indicated in Fig. 4, efficiency curves for R-113 and R-123 intersect at ~30 ℃; below 30 ℃ R–123 has a slightly higher efficiency than R-113 and a reverse performance occurs when temperature is above 30℃. The intersection of efficiency curves of R–123 and R–113 can be explained by their thermophysical properties. Since R–123 has higher values of thermal conductivity and latent heat than those of R–113, it means that R–123 has a better heat transfer performance than that of R-113. On the other hand, R–123 has a higher specific heat than that of R–113. Since a regenerator is not considered in this study, the inlet condition of the working fluid would become superheated before entering the condenser. This imposes a higher load on the condenser and reduces the system efficiency as a dry fluid goes through an isentropic expansion at the exit of the turbine. Furthermore, the latent heat ‘‘bandwidth’’ at s low pressure on the T-s diagram also affects the system efficiency. A broader latent heat bandwidth represents a greater amount of heat must be taken away from the condenser. This effect is not significant since the temperature is very low as the condenser is operated under a low pressure. However, this effect becomes important as the inlet temperature of the turbine increases. Therefore, a careful selection of the working fluid under different working temperatures is crucial in optimizing the performances of ORCs.

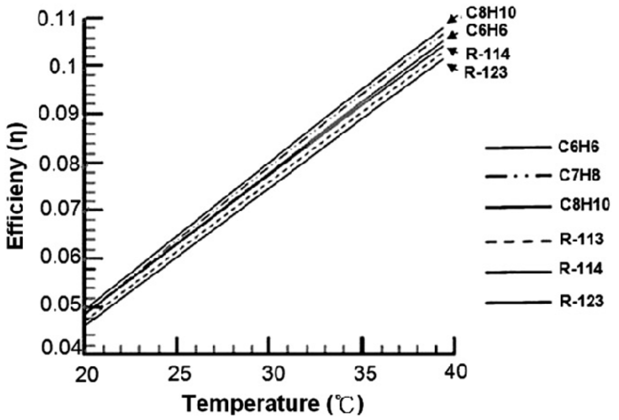

Fig. 4 Case 1: System efficiency-turbine inlet temperature dependence for dry fluids

As for the wet fluids, both R-11 and R-12 have a rather better performance in system efficiency than that of the other wet fluids except for R-152[a] as shown in Fig. 5. This is because that both R-11 and R-12 are nearly identical to isentropic wet fluids, i.e., their saturation vapor curves are virtually vertical in this region. In practical operation with a careful arrangement, the fluid conditions at the turbine exit can be adjusted to fall on the saturation vapor curve. This has two obvious advantages: no occurrence of moisture during isentropic expansion in the turbine, and no need for using a regenerator to reduce condensation load.

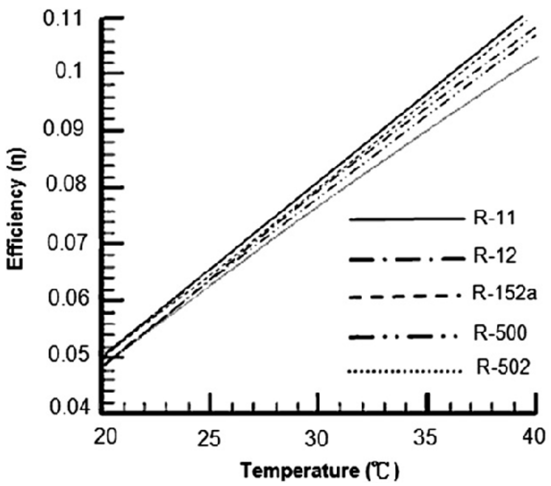

Fig. 5. Case 1: System efficiency-turbine inlet temperature dependence for wet fluids

Based on the analysis stated above, three factors are believed to have major impacts on the system efficiency according to the thermophysical properties of the fluids under investigation: the slope of saturation curve, specific heat, and latent heat. Under a broader range of working temperature between T_H and T_L, latent heat can vary significantly. It has a direct influence on the energy absorbed in the evaporator, and the "bandwidth" of the saturation curve in the T-s diagram at the low temperature end; and consequentially, the net work done by the cycle.

As stated above, isentropic fluids in general have a better performance in efficiency. Wet fluids also perform well in system efficiency. However, a major problem with the wet fluids is the possible damage to the turbine blades due to corrosion from the moisture of the two-phase state of the working fluids as they do not follow a saturation curve during expansion in the turbine. Therefore, emphasis is also focused on the influence of the "quality" of the wet fluids on turbine efficiency. Here, the working conditions are: 5℃ for the low temperature end, 40℃ for the high temperature end, and a saturation state for the turbine inlet. Figure 6 shows the dependence of the system efficiency and turbine efficiency on the quality of the fluids at turbine exit. As the exit quality reaches 1, i.e., a high-efficiency turbine is not necessary, R-152[a] has the lowest system efficiency of 6.7 % while R-11 and C_7H_8 have the highest system efficiency around 8.5 %. An isentropic fluid such as R-11 expands in the turbine isentropically to a low pressure with a quality very close to 1. Therefore, design constraints are less stringent for such fluids. When the quality at the turbine exit drops, turbine efficiency must be reduced accordingly in order to have an exit quality close to 1. R-152[a] on the other hand has a rather slowly-varying positive slope of the saturation vapor curve; this results in a very low quality at turbine exit. The situation can be improved by operating the fluid in super-heated region before it entering the turbine.

In practice, it can be designed that the fluid at turbine exit is saturated vapor in order for R-152[a] to operate under a low temperature range.

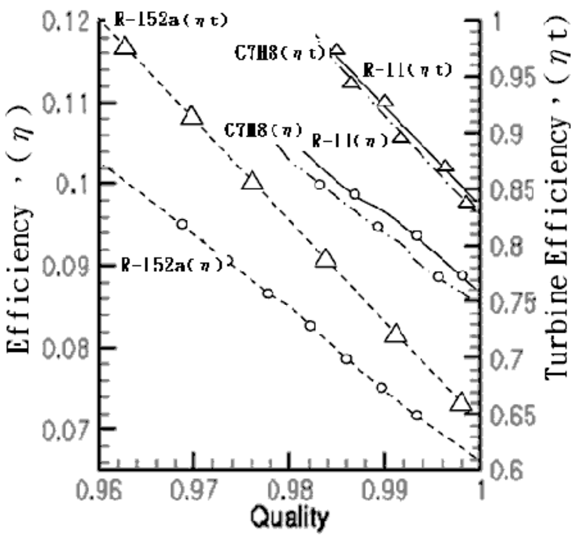

Fig. 6 Case 1: System Efficiency and Turbine Efficiency on Quality

As stated above, heat transfer rates in the evaporator and condenser are the key contributions to the system irreversibility. The following calculations are based on a 10 MW heat source. The irreversibility in a system differs from component to component. Figure 7 shows the irreversibilities contributed from the major components of an ORC under various turbine inlet pressures for the fluid C_7H_8. It shows that the condenser has the major contribution of irreversibility; and thus, improvement of the condenser performance is an important concern when the upper end of the cycle is operated under a high pressure. Figure 8 shows the irreversibilities for various working fluids under a wide range of turbine inlet pressures at a constant turbine inlet temperature. It is clear that the enthalpies of the fluids at point 3, i.e., the turbine inlet, increase as the pressure increases, and so do their respective irreversibilities.

For the sake of a better discussion, the availability ratio defined in Eq. (13) is also used as a guideline in selecting a suitable working fluid. Figure 9 shows the variations of the system efficiency and availability ratio versus turbine inlet pressure for the fluid C_8H_{10}. As expected, the availability ratio decreases as the turbine inlet pressure increases. Therefore, an optimal choice of the working fluid may be achieved by taking the intersection of the efficiency curve and the availability ratio curve. From Fig. 9, it seems that a system pressure below 500 kPa is adequate for recovering low-grade energy if C_8H_{10} is used as the working fluid.

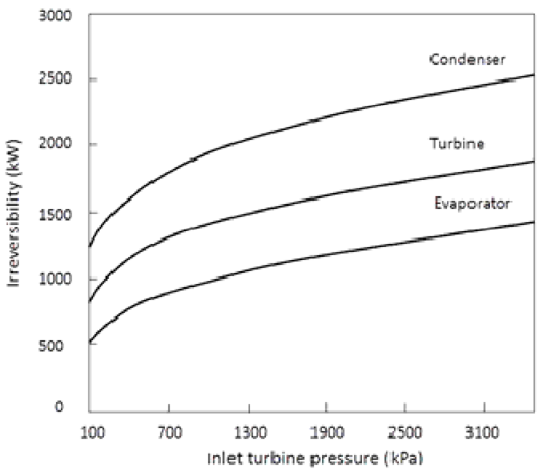

Fig. 7 Distributions of Irreversibilities in the Major Components of a Rankine cycle for Fluid C_7H_8 at $T_H = T_3 + 15$ ℃

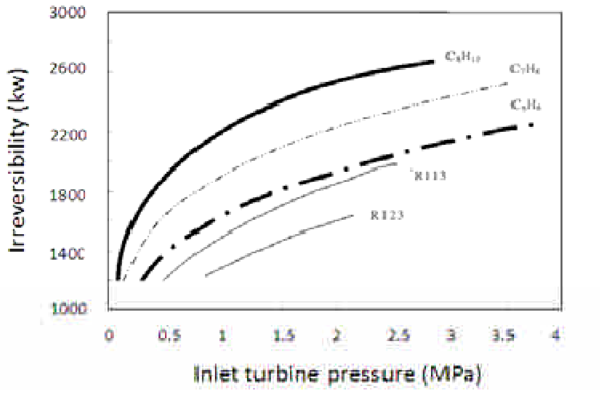

Fig. 8 Irreversibility versus Turbine Inlet Pressure at $T_H = T_3 + 15$ ℃

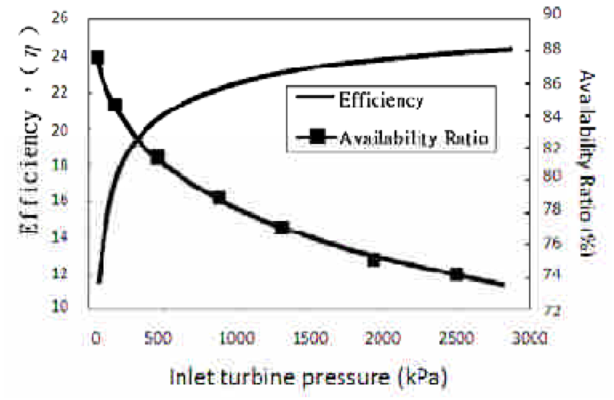

Fig. 9 System Efficiency and Availability Ratio versus Turbine Inlet temperature for fluid C_8H_{10}

4.2 Case 2: Solar energy as high-temperature reservoir

In this case the energy source of an ORC system is supplied by solar energy in the evaporator. The high temperature source may be from a solar pond or a solar collector. As a scoping calculation, the operation conditions of the ORC are: 20℃ for pump inlet temperature and 40 to 60℃ for turbine inlet temperature. The study shows that benzene-series fluids have a better performance in efficiency than that of the refrigerant-series fluids. Among the wet fluids, R-11 has the best performance. This result is similar to Case 1. Figure 10 shows the dependences of the system efficiency and turbine efficiency on turbine exit quality. Compared with Case 1, the difference is the turbine exit quality; resulting from different slopes of the saturation vapor curves at a new temperature range (between T_H and T_L) . A suitable working fluid can thus be selected based on a consideration of the system efficiency in order to meet the required efficiency of the turbine.

A composite system which combines OTEC and solar energy to serve as an ORC system with a higher temperature difference between T_H and T_L, i.e., an evaporator with a higher operation temperature supplied by solar energy and a condenser cooled by cold water in deep sea, is believed to yield an even better performance. In this case, the system efficiency can be as high as 11 to 15 % if the operation temperature of the evaporator is between 40 and 60℃. As shown in Fig. 16, isentropic (or nearly isentropic) fluids have higher efficiencies. By comparing R-11 and R-113, R-11 has a better efficiency performance than that of R-113. This can be explained from the T-s diagram in which R-113, a dry fluid, becomes superheated at the turbine exit, and this reduces the area of net work in the T-s diagram. Furthermore, R-11 has a higher thermal conductivity and latent heat than those of R-113. Therefore, an optimal operation of high efficiency can be achieved with a proper selection of the working fluid in the combined system.

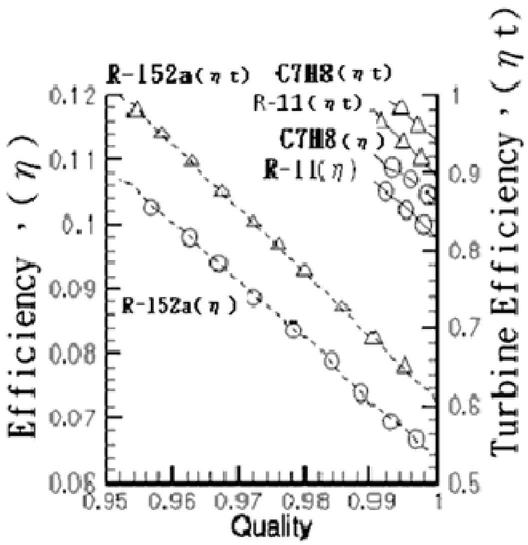

Fig. 10 Case 2: System Efficiency and Turbine Efficiency on Quality

5. CONCLUSION

System efficiency can be optimized by selecting a proper working fluid operated at suitable working conditions. In terms of disadvantages, dry fluids in general generate superheated vapor at the turbine exit, and this reduces the area of net work in the T-s diagram. A generator may be needed in order to relieve the cooling load of the condenser. The major disadvantage for the wet fluids is their moisture content during expansion process in the turbine. Isentropic (or nearly isentropic) fluids in general is free from the concern of the moisture content, and does not need a regenerator to relieve the cooling load of the condenser. The only issues of concern for the isentropic fluids are their cost, chemical stability, and safety. Therefore, they are considered to be the best candidates of the working fluids for ORCs. A combination of OTEC and solar energy to yield a higher temperature difference in an ORC system is believed to give an even better performance in efficiency.

NOMENCLATURE

c_p = specific heat
h = enthalpy
h_{4a} = turbine exit enthalpy for an irreversible process
T = temperature
I = irreversibility rate
v = specific volume
w_{12} = work done by pump
w_{34a} = work done by turbine for an irreversible process
h_{fg} = latent heat
k = thermal conductivity
m = mass flow rate
p = pressure
q_{23} = heat added to evaporator
s = entropy
x = quality
η_{th} = thermal efficiency
w_{34} = work done by turbine
φ = availability ratio

Subscripts
1~4 locations of states
o ambient
sys system
t turbine
H High
L Low
tot total

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無研發成果推廣資料

98 年度專題研究計畫研究成果彙整表

計畫主持人：王曉剛	計畫編號：98-2221-E-214-059-
計畫名稱：結合太陽能與海洋溫差於有機朗肯循環發電研究	

成果項目			量化			單位	備註（質化說明：如數個計畫共同成果、成果列為該期刊之封面故事...等）
			實際已達成數（被接受或已發表）	預期總達成數(含實際已達成數)	本計畫實際貢獻百分比		
國內	論文著作	期刊論文	0	0	100%	篇	
		研究報告/技術報告	0	0	100%		
		研討會論文	0	0	100%		
		專書	0	0	100%		
	專利	申請中件數	0	0	100%	件	
		已獲得件數	0	0	100%		
	技術移轉	件數	0	0	100%	件	
		權利金	0	0	100%	千元	
	參與計畫人力（本國籍）	碩士生	2	0	100%	人次	
		博士生	0	0	100%		
		博士後研究員	0	0	100%		
		專任助理	0	0	100%		
國外	論文著作	期刊論文	1	0	100%	篇	
		研究報告/技術報告	0	0	100%		
		研討會論文	1	0	100%		
		專書	0	0	100%	章/本	
	專利	申請中件數	0	0	100%	件	
		已獲得件數	0	0	100%		
	技術移轉	件數	0	0	100%	件	
		權利金	0	0	100%	千元	
	參與計畫人力（外國籍）	碩士生	0	0	100%	人次	
		博士生	0	0	100%		
		博士後研究員	0	0	100%		
		專任助理	0	0	100%		

其他成果 (無法以量化表達之成果如辦理學術活動、獲得獎項、重要國際合作、研究成果國際影響力及其他協助產業技術發展之具體效益事項等，請以文字敘述填列。)	無

	成果項目	量化	名稱或內容性質簡述
科教處計畫加填項目	測驗工具(含質性與量性)	0	
	課程/模組	0	
	電腦及網路系統或工具	0	
	教材	0	
	舉辦之活動/競賽	0	
	研討會/工作坊	0	
	電子報、網站	0	
	計畫成果推廣之參與（閱聽）人數	0	

國科會補助專題研究計畫成果報告自評表

請就研究內容與原計畫相符程度、達成預期目標情況、研究成果之學術或應用價值（簡要敘述成果所代表之意義、價值、影響或進一步發展之可能性）、是否適合在學術期刊發表或申請專利、主要發現或其他有關價值等，作一綜合評估。

1. 請就研究內容與原計畫相符程度、達成預期目標情況作一綜合評估 ■達成目標 □未達成目標（請說明，以 100 字為限） □實驗失敗 □因故實驗中斷 □其他原因 說明：
2. 研究成果在學術期刊發表或申請專利等情形： 論文：■已發表 □未發表之文稿 □撰寫中 □無 專利：□已獲得 □申請中 ■無 技轉：□已技轉 □洽談中 ■無 其他：（以 100 字為限）
3. 請依學術成就、技術創新、社會影響等方面，評估研究成果之學術或應用價值（簡要敘述成果所代表之意義、價值、影響或進一步發展之可能性）（以 500 字為限） 本研究目前完成了所選擇之十一種有機工作流體的分析模式，分別應用於不同熱源，海洋溫差與太陽能熱源。經由熱力分析與各類有機流體之物性來判斷。在本研究兩案例中，適當的工作流體選擇與各流體的飽和曲線特性，以及在不同熱源範圍下，從效率圖上可以發現每個流體在其適當的熱源下，可以得到其有效的使用範圍。同時也可得知在實際運轉時，針對系統的設備來進行改善，對系統之整體效率而言，是一大幫助。乾流體於本研究的缺點在於會使渦輪機操作出口落點處於過熱蒸汽，造成 T-s 圖上的缺角而使效率沒有提升。何況過熱蒸汽將使冷凝器太過於負擔而需使用再生器來改善，但設備成本也因此較高；濕流體的缺點就在於液滴的產生，將破壞渦輪機；近似等熵的濕流體其較無渦輪機破壞之顧慮，也無須像乾流體多裝再生器來減少冷凝器的負擔。此研究已理論證實結合太陽能與海洋溫差於有機朗肯循環發電之可行性，其效率雖然尚低，然若能結合太陽能溫水池當作高溫端，並對定發電功率下之流體，針對其管路設計以使壓力降與熱傳率達到最佳化，則海洋能發電之效率更能提高.