



**Programa de las  
Naciones Unidas  
para el Medio Ambiente**



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COMITÉ EJECUTIVO DEL FONDO MULTILATERAL  
PARA LA APLICACIÓN DEL  
PROTOCOLO DE MONTREAL  
Sexagésima sexta Reunión  
Montreal, 16 – 20 de abril de 2012

**COSTOS ADICIONALES RELACIONADOS CON LAS MODIFICACIONES  
PARA FABRICAR INTERCAMBIADORES DE CALOR  
(DECISIONES 61/45 Y 62/61)**

1. Durante la preparación de la 59ª reunión del Comité Ejecutivo, la Secretaría identificó una cuestión relacionada con el costo de la conversión del componente de fabricación respecto de los costos adicionales de explotación. En su Decisión 59/14, el Comité Ejecutivo decidió aplazar el examen hasta la 60ª reunión y, posteriormente, en su Decisión 60/45, aplazarlo hasta la 61ª reunión. En la 61ª reunión, se examinaron y resolvieron las cuestiones planteadas en el documento UNEP/OzL.Pro/ExCom/61/51, excepto para la cuestión de la financiación para la reconversión de líneas de fabricación de intercambiadores de calor.

2. El Comité Ejecutivo tomó la Decisión 61/45 c) en la que se pide a la Secretaría que prepare un documento acerca del nivel de costos adicionales relacionados con el equipamiento de herramientas para fabricar intercambiadores de calor de aleta y placa, para que sea considerado por el Comité Ejecutivo en la 62ª Reunión. La Secretaría preparó el documento UNEP/OzL.Pro/ExCom/62/55 que figura en el Anexo I al presente documento. Durante las deliberaciones en el Comité Ejecutivo, se cuestionó la calidad del documento dado que, a juicio de un delegado, no tomaba debidamente en cuenta las cuestiones muy técnicas que se plantean. En su Decisión 62/61, el Comité Ejecutivo pidió a la Secretaría del Fondo que prepare un nuevo documento sobre los costos adicionales relacionados a la adaptación de líneas de fabricación de intercambiadores de calor como ayuda para sus deliberaciones, incorporando las opiniones que pudiese recibir de expertos, los organismos de ejecución y los miembros del Comité Ejecutivo. La Secretaría emitió el documento UNEP/OzL.Pro/ExCom/65/53 en respuesta a la decisión 62/61, tomando en cuenta las cuestiones de índole altamente técnica planteadas. El Comité Ejecutivo comenzó a deliberar sobre el documento pero, tras las deliberaciones iniciales, acordó diferir hasta la 66ª reunión el tratamiento de los costos adicionales relacionados con las modificaciones para la fabricación de intercambiadores de calor.

3. La Secretaría emite nuevamente el documento preparado para la 65ª reunión, agregando las observaciones técnicas proporcionadas por el Bando Mundial acerca de la cuestión. Dichas observaciones se presentan en el Apéndice III de este documento.

#### Enfoque

4. La Secretaría ha analizado los debates mantenidos en la 62ª reunión y ha realizado una transcripción de las partes de la intervención de un miembro del Comité Ejecutivo relacionadas con cuestiones técnicas, en las que se abordan aspectos específicos para la investigación de diversos asuntos concretos. La Secretaría también ha recibido una carta del departamento de desarrollo de Petra Engineering Industries Co. en Jordania, beneficiaria de un proyecto de demostración para la reconversión de líneas de fabricación de aire acondicionado a la tecnología HFC-410A, y para la que hay pendiente una decisión sobre la financiación de la reconversión de intercambiadores de calor. La información correspondiente figura en los Apéndices I y II, respectivamente. La Secretaría ha mantenido discusiones con los organismos de ejecución sobre este particular, sobre todo con el PNUD durante el examen de los proyectos de reconversión en el marco del plan de gestión de eliminación de HCFC para Indonesia. La Secretaría ha analizado las contribuciones conexas, ha preparado una lista de cuestiones técnicas que se han de examinar y ha llegado a la conclusión de que los trabajos afines debe realizarlos un experto externo. La Secretaría identificó un experto adecuado y se puso en contacto con él para encargarle un estudio. El experto se ha jubilado a principios de este año, es miembro del Comité de Opciones Técnicas - Refrigeración, y ya ha sido director técnico para el desarrollo de enfriadores y jefe de tecnología y desarrollo de compresores en una importante empresa de fabricación de aire acondicionado. El estudio mencionado figura en el Apéndice I al presente documento.

5. El estudio técnico gira en torno a las prácticas en el sector de fabricación de aire acondicionado durante la reconversión de HCFC-22 a HFC-410A. Quizá valga la pena destacar algunos aspectos en esta nota introductoria:

- a) El experto señaló que la cuestión de la presión del sistema y el aumento necesario de la anchura de las paredes de la tubería tienen que ver con las posibles temperaturas elevadas y la consecuente presión que experimentan durante el transporte, es decir, es independiente del clima en el lugar de utilización del producto y un nivel de presión más elevado que en condiciones de funcionamiento;
- b) Otro punto que señaló el experto era que la resistencia a la presión se estableció, en cualquier caso, como múltiplo de la presión más alta probable, definida con arreglo a la temperatura descrita en el párrafo a) precedente;
- c) También destacó que en respuesta a las dificultades que entraña la reconversión de HCFC-22 a HFC-410A, la industria ha cambiado de procedimiento para garantizar una resistencia a la presión determinada pasando de un método estático, según el cual cabe prever cinco veces la presión máxima, a un método dinámico más complejo, que se acerca más a la realidad, en el que las presiones de prueba se limitan a tres veces la presión más probable, pero el sistema tiene que soportar primero 200 000 ciclos en los que se aplica primero la presión más elevada probable y luego se relaja. El experto explicó que este procedimiento aplica una tensión adicional considerable al equipo y muestra varias debilidades que no se manifestaban con la prueba de presión estática, pese a que la presión más alta en la prueba dinámica es inferior a la de la prueba estática; y
- d) Como consecuencia, los sistemas de aire acondicionado en general, y los intercambiadores de calor en particular, se diseñan con características al menos igual de buenas, y posiblemente mejores, que obliga a introducir algunos pequeños cambios de diseño, pero sin llegar a necesitar modificaciones importantes.

6. El consultor cuantificó además los posibles costos que conllevarían las pruebas dinámicas y las herramientas necesarias para garantizar un producto de suficiente calidad. En el estudio también se explica en detalle diversas modificaciones típicas del diseño, explicaciones que son fruto de una experiencia práctica obtenida con dificultad y que, por ende, quizá sirva a los organismos de ejecución y a las empresas de orientación sobre el rediseño al realizar la reconversión. La Secretaría considera que el estudio abarca, directa o indirectamente, todas las cuestiones técnicas planteadas en las deliberaciones del Comité Ejecutivo o con la Secretaría.

7. En el Cuadro 1 se muestran los costos adicionales estimados que conllevaría la reconversión de intercambiadores de calor.

Cuadro 1: Costo de la reconversión de intercambiadores de calor

| <b>Elemento de costo</b> | <b>Costo para empresas medianas (\$EUA)</b> | <b>Costo para pequeñas empresas (\$EUA)</b> |
|--------------------------|---|---|
| Herramientas             | 15 000                                      | 15 000                                      |
| Equipo de prueba         | 100 000                                     | Depende del caso particular                 |
| Imprevistos              | 11 500                                      | Depende del caso particular                 |
| Total (por empresa)      | 126 500                                     | ≥15 000                                     |

8. La Secretaría discutió con el consultor las conclusiones que pueden referirse del estudio, en lo que respecta a la ayuda que necesitarían los fabricantes en países que operan al amparo del Artículo 5. Se llegó a la conclusión de que la reconversión a HFC-410A requiere generalmente mucho trabajo de ingeniería, no tanto por la necesidad de reconvertir la línea de fabricación, sino para adoptar pequeños cambios de diseño que permitan un comportamiento dinámico diferente del refrigerante a lo largo del ciclo; si no se efectúan esos cambios suelen aparecer con frecuencia vibraciones, que llegan a romper las soldaduras, etc. Dado que las soldaduras son especialmente numerosas en los intercambiadores de calor, las pruebas dinámicas garantizan la solidez del intercambiador de calor, así como la calidad general de la fabricación.

9. A juicio del consultor, las empresas medianas, tales como Petra Engineering Industries Co., podrían efectuar las pruebas dinámicas por sí mismas. Estos fabricantes de tamaño medio se caracterizarían por tener un consumo superior a 100 toneladas métricas de HCFC-22 y grupos de control de calidad y certificación separados dentro de la propia empresa. En el caso de los fabricantes pequeños, sería considerablemente más adecuado crear un centro competente en el país y permitir a las empresas el acceso al centro para realizar pruebas y adquirir conocimientos prácticos. El consultor estimó que el enfoque centralizado tiene más sentido, habida cuenta de las competencias necesarias para realizar las pruebas. Los fabricantes pequeños no necesitan llevar a cabo un número de pruebas tal que justifique disponer de equipos para tales fines, ni tienen la oportunidad de hacer un número de pruebas suficiente para adquirir conocimientos prácticos por sí mismos. Por consiguiente, el consultor recomienda, en los casos en los que los fabricantes producen sus propios intercambiadores de calor, que se ayude a las empresas medianas y grandes con equipos de pruebas y financiación limitada para mejorar sus herramientas de fabricación. Los países que cuenten con muchas pequeñas empresas se les ofrecería financiación para crear un centro de pruebas especializado, así como herramientas para ayudar a los pequeños fabricantes, y para las empresas pequeñas sin acceso a dicho centro ofrecerles financiación para comprar equipos pequeños de prueba con el fin de probar los equipos lo mejor posible dentro de sus posibilidades.

10. El consultor subrayó que la fabricación de intercambiadores de calor de un solo modelo y a gran escala requeriría herramientas diferentes, dependiendo de la flexibilidad exacta de las herramientas utilizadas en el proceso, o la falta de dicha flexibilidad. Este aspecto no pudo evaluarse en el estudio, ya que depende sobremanera de la configuración específica. Ahora bien, aparentemente el único caso en el que la fabricación a gran escala podría suponer un problema en instalaciones que no son propiedad de multinacionales sería el de algunos fabricantes en China, donde el nivel de financiación hasta 2015 ya se ha acordado. Por consiguiente, no hay necesidad de tratar de inmediato este caso particular.

11. El Comité Ejecutivo decidió, en el caso de varias comunicaciones, determinar el costo máximo admisible de la reconversión de intercambiadores de calor, pero retener la financiación hasta tanto no se haya resuelto el problema general de la financiación de dicha reconversión. En el Cuadro 2 se indican los proyectos afectados y la cantidad asignada.

**Cuadro 2: Proyectos del plan de gestión de eliminación de HCFC cuyas financiaciones para la reconversión de intercambiadores de calor aún no se ha aprobado**

| <b>Título</b>  | <b>Organismo de ejecución</b> | <b>Decisión</b> | <b>Costos aprobados (\$EUA)</b> | <b>Costo adicional solicitado para la reconversión de intercambiadores de calor (\$EUA)</b> |
|--|-------------------------------|-----------------|---------------------------------|---|
| Eliminación del HCFC-22 y del HCFC-141b de la fabricación de equipos unitarios de aire acondicionado en Petra Engineering Industries Co. | UNIDO                         | 60/41*          | 2 167 033                       | 737 000   |
| Indonesia: Plan de gestión de eliminación de HCFC (etapa I) (plan del sector de aire acondicionado)                                      | UNDP                          | 64/42**         | 1 995 519                       | 52 800  |

\* Párrafo b) de la Decisión: El Comité Ejecutivo decidió examinar la financiación de las partidas de costos conexas a la conversión de la fabricación de termointercambiadores aleteados, que quedaron excluidos de financiación de conformidad con el apartado a) antedicho, cuando se analice la cuestión de criterios conexas que se recoge en el documento UNEP/OzL.Pro/ExCom/60/47.

\*\* Párrafo h) de la Decisión: El Comité Ejecutivo decidió pedir al PNUD, en su calidad de organismo director, que presente una solicitud de financiación para la conversión del intercambiador de calor cual componente del primer tramo del plan de gestión de eliminación de HCFC para Indonesia, una vez que el Comité Ejecutivo haya tomado una decisión sobre la financiación de la conversión para la fabricación del intercambiador de calor, en la inteligencia de que el nivel de financiación se calcularía sobre la base de dicha decisión y que no superaría un máximo de 52 800 \$EUA según corresponde

12. A diferencia de lo que se esperaba cuando se adoptaron las decisiones anteriores, la respuesta técnica del estudio no estipula si el cambio de herramientas es o no necesario. Si bien teóricamente no se requieren herramientas adicionales para efectuar la reconversión, el consultor las estima necesario para crear instalaciones de prueba. La Secretaría calculó, basándose en el estudio, los niveles de costos que se indican en el Cuadro 3 para los dos proyectos concernidos.

**Cuadro 3: Nivel de financiación propuesto para los proyectos de reconversión de intercambiadores de calor**

| <b>Título</b>  | <b>Nivel de financiación propuesto (\$EUA)</b> |
|--|--|
| Eliminación del HCFC-22 y del HCFC-141b de la fabricación de equipos unitarios de aire acondicionado en Petra Engineering Industries Co. | 126 500  |
| Indonesia: Plan de gestión de eliminación de HCFC (etapa I) (plan del sector de aire acondicionado)                                      | 52 800   |

#### Recomendación de la Secretaría

13. Habida cuenta del análisis anterior, del contenido del documento UNEP/OzL.Pro/ExCom/62/55 incluido en el Anexo I al presente documento, y de la información facilitada en el estudio que figura en el Apéndice I, el Comité Ejecutivo pudiera:

- a) Considerar que en casos en los que se efectúe la reconversión de un fabricante de productos de aire acondicionado a una tecnología alternativa con una presión de trabajo considerablemente más elevada, y cuando este fabricante disponga de sus propias instalaciones de fabricación de intercambiadores de calor, los gastos en concepto de pruebas dinámicas de los intercambiadores de calor forman parte del costo de la reconversión a fin de garantizar que son lo suficientemente resistentes a la presión;

- b) Solicitar a la Secretaría que utilice, para esos casos, la información técnica facilitada en el documento UNEP/OzL.Pro/ExCom/66/51 y el estudio técnico conexo, así como cualquier otra información pertinente disponible para determinar en qué medida dichos costos son admisibles y adicionales;
- c) Aprobar, para el proyecto “Eliminación del HCFC-22 y del HCFC-141b de la fabricación de equipos unitarios de aire acondicionado en Petra Engineering Industries Co.” (Jordania), una financiación adicional de 126 500 \$EUA más unos costos de apoyo para la ONUDI de 9 488 \$EUA, con el fin de sufragar los costos que conllevan las modificaciones en la producción y prueba de intercambiadores de calor, de conformidad con la Decisión 60/41; y
- d) Aprobar, para la etapa I del plan de gestión de eliminación de HCFC para Indonesia (plan del sector de aire acondicionado), una financiación adicional de 52 800 \$EUA más unos costos de apoyo para el PNUD de 3 960 \$EUA, con el fin de sufragar los costos que conllevan las modificaciones en la producción y prueba de intercambiadores de calor, de conformidad con la Decisión 64/42, y solicitar a la Secretaría que actualice el Acuerdo con Indonesia en consecuencia cuando introduzca cualquier cambio relativo a la determinación del consumo básico de referencia para Indonesia.

**ANNEX/ ATTACHMENTS FOR DOCUMENT UNEP/OzL.Pro/ExCom/66/51**

- ANNEX I DOCUMENT UNEP/OzL.Pro/ExCom/62/55
- ATTACHMENT I: A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR CONVERSION FROM R-22 TO R-410A
- ATTACHMENT II VIEWS EXPRESSED RELATED TO RETOOLING FOR MANUFACTURING HEAT EXCHANGERS
- ANNEX I: EXCERPT FROM THE CONTRIBUTION OF ONE DELEGATE TO THE DISCUSSION DURING THE 62ND MEETING OF THE EXECUTIVE COMMITTEE (TRANSCRIPT)
  - ANNEX II: “ R410A/R22 COMPARISON OF PRESSURE LEVELS”, SUBMITTED BY PETRA ENGINEERING R&D LABS
- ATTACHMENT III COMMENTS ON “A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR CONVERSION FROM R-22 TO R-410A” SUBMITTED BY THE WORLD BANK



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PARA LA APLICACIÓN DEL  
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Sexagésima segunda Reunión  
Montreal, 29 de noviembre al 3 de diciembre de 2010

**COSTOS ADICIONALES  
RELACIONADOS CON LAS MODIFICACIONES PARA FABRICAR  
INTERCAMBIADORES DE CALOR  
(DECISIÓN 61/45)**

Los documentos previos al período de sesiones del Comité Ejecutivo del Fondo Multilateral para la Aplicación del Protocolo de Montreal no van en perjuicio de cualquier decisión que el Comité Ejecutivo pudiera adoptar después de la emisión de los mismos.



1. En los preparativos de la 59ª Reunión del Comité Ejecutivo la Secretaría determinó una cuestión conexas al costo de convertir la fabricación de componentes en comparación con el costo adicional de explotación. Sirviéndose de su decisión 59/14 el Comité Ejecutivo decidió posponer el examen de dicha cuestión hasta la 60ª Reunión y, posteriormente, en su decisión 60/45, posponerlo hasta la 61ª Reunión.

2. Se debatieron y solventaron las cuestiones planteadas en el documento conexo UNEP/OzL.Pro/ExCom/61/51 de la 61ª Reunión, salvo en lo tocante a la cuestión de los fondos de financiación necesarios para la conversión de la fabricación de intercambiadores de calor. En consecuencia, el Comité Ejecutivo tomó la decisión 61/45 c), por la que se pidió a la Secretaría que preparara un documento, basándose en las secciones pertinentes del documento UNEP/OzL.Pro/ExCom/61/51, que tratara de la cuantía de los costos adicionales conexos al equipamiento de herramientas para la fabricación de intercambiadores de calor aleteados, con objeto de ser examinados en la 62ª Reunión del Comité Ejecutivo.

#### Costo adicional de los intercambiadores de calor

3. En el desempeño de sus tareas de examen de las presentaciones de proyectos, la Secretaría determinó que el hecho de si debía considerarse como un costo adicional la conversión de la fabricación de intercambiadores de calor constituía una cuestión. En los párrafos que se indican *infra* se intenta explicar los sumamente técnicos aspectos del tema para posibilitar al Comité Ejecutivo a tomar a su respecto una decisión con conocimiento de causa.

4. La fabricación de los intercambiadores de calor a la que se hace mención en el presente documento se refiere a la fabricación de termointercambiadores cuya función es la de transferir al aire el calor almacenado en el refrigerante. Tales intercambiadores constan, por lo general, de un cierto número de tubos de cobre dotados de aletas planas perpendiculares a los mismos.

5. La fabricación de las aletas se lleva a cabo con complejos troqueles con los que se martillean repetidamente las aletas hasta conformarlas. El diámetro externo de los tubos es mínimo en comparación con los agujeros internos de las aletas, de forma que éstas puedan alinearse sin dificultad en una serie de tubos. Lo normal es que los tubos hayan sido enderezados previamente y después doblados en U formando horquillas (“horquillas huecas”), de manera que cada tubo pase dos veces por el intercambiador de calor; cada intercambiador de calor puede llevar gran número de tubos. Estos tubos se entrelazan formando una pila de aletas (de hasta varios cientos) en una bancada horizontal. Una vez emplazadas todas las horquillas huecas en la pila de aletas, por el interior de cada tubo se pasa una varilla dotada con una punta de bola de precisión ligeramente mayor que el diámetro interior de los tubos, agrandando así ligeramente éste y, por ende, su diámetro exterior, lo que da lugar a que se produzca un encaje entre el tubo propiamente dicho y la aleta correspondiente. Los tubos se expanden al mismo tiempo que se ejecuta su proceso de fabricación, el cual tiene lugar a gran velocidad. De tratarse de una tirada de baja producción, hay veces en que los tubos se expanden uno a uno. Estos intercambiadores se denominan intercambiadores de calor (o termointercambiadores) de aletas (o aleteados).

6. Los intercambiadores de calor que transfieren éste desde un refrigerante al aire son algo muy común y corriente en los sistemas de refrigeración y de aire acondicionado, especialmente en el caso de la fabricación de sistemas a gran escala. En éste último caso los intercambiadores de calor se optimizan para cada modelo y se compran a un proveedor externo o, lo que es más frecuente, se optimizan teniendo presente la gama de modelos del fabricante y, además, se fabrican in situ. Por lo general se usa el mismo diámetro externo de tubo para equipos de una amplia gama de capacidades. Estos intercambiadores de calor no difieren grandemente en su diseño ni en los materiales utilizados cuando se compara la tecnología HCFC-22 con sus diversas alternativas actuales (salvo en el caso del amoníaco y el CO<sub>2</sub>).

7. En opinión de los peritos consultados por la Secretaría, no es necesario reducir el diámetro del tubo, desde el punto de vista del desempeño del sistema, cuando se pase de HCFC-22 a HFC-410A o HFC-32; lo que también es aplicable cuando se pasa a HFC-407C y cuando se utilizan los hidrocarburos HC-290 y HC-1270. No obstante, sí es necesario ajustar ligeramente el grosor de la pared del tubo a fin de incrementar su resistencia ante las presiones de trabajo, las cuales son más elevadas en el caso del HFC-410A y del HFC-32. Este enfoque conlleva una inversión de capital en equipos que es considerablemente menor, por lo que es el preferido de los fabricantes para la fase inicial de la conversión. Otra alternativa es emplear en la fabricación de los tubos unos tipos de cobre específicos, más costosos, que les hace más resistentes a la presión con las mismas dimensiones, o bien una combinación de ambos tipos de cobre. La reducción del diámetro externo del tubo, como se ha pedido en algunas propuestas de proyecto, se beneficia de ciertas ventajas que resultan de las propiedades del HFC-410A y que no presenta el HCFC-22, todo lo que deriva en una reducción al mínimo del sistema, además de reducciones de peso y de costos. El ahorro en costos es tan considerable que pueden incluso fabricarse termointercambiadores de calidad superior reduciendo considerablemente además los costes de explotación. Las propuestas de proyecto recibidas por la Secretaría indican que los ahorros adicionales pueden ser superiores a los 2 \$EUA/kg de HCFC-22.

8. Sin embargo, como ya se puso de manifiesto en las presentaciones de otros proyectos, los costos del equipamiento de herramientas para la fabricación de tubo de diámetro externo pequeño son muy considerables. En el caso de los sistemas intercambiadores de calor que funcionen con CO<sub>2</sub> probablemente se necesitaría utilizar tubos de menor diámetro dadas las muy altas presiones de trabajo de los sistemas de funcionamiento por CO<sub>2</sub> y su diferente capacidad por volumen. Los sistemas que empleen refrigerantes que sean inflamables (hidrocarburos y, en menor medida, HFC-152a y HFC-32), permiten reducir el volumen de refrigerante que hay que cargar al utilizar tubos de menor diámetro, lo que permite utilizar refrigerantes inflamables en tales equipos de refrigeración con muy poco incremento de las medidas de seguridad en comparación con los actuales sistemas que utilizan HCFC.

9. Las máquinas que se utilizan para la fabricación de los intercambiadores de calor se personalizan, hasta cierto punto, lo que se hace concretamente en lo tocante a las dimensiones externas del tubo. Un cambio de las dimensiones externas del tubo conllevará tener que cambiar el equipo, especialmente los troqueles con los que se fabrican las aletas, las máquinas para horquillar los tubos de cobre, los equipos de soldadura fuerte automatizada y las máquinas empleadas para expandir los tubos. En lo que a la fabricación de productos de consumo se refiere (climatizadores de habitación, etc.), se emplean por lo general máquinas de precisión totalmente automatizadas cuyos costos de recambio o de modificación son relativamente elevados. Los productos de aplicación comercial e industrial se fabrican frecuentemente con un grado de automatización inferior a fin de poder atender a los diseños específicos del cliente. No obstante, las máquinas empleadas requieren también un alto grado de precisión.

10. Las empresas actualmente acostumbran fabricar en sus instalaciones sus propios intercambiadores de calor, lo que les permite un mayor grado de flexibilidad en el proyecto y fabricación de sistemas de aire acondicionado de gran envergadura siguiendo las especificaciones del cliente, pudiendo quizás obtenerse también una reducción de los costos de explotación. En la inmensa mayoría de los casos presentados a la atención de la Secretaría, los fabricantes de equipos de refrigeración y de aire acondicionado produjeron en sus instalaciones sus propios intercambiadores de calor.

11. Tras la 61<sup>a</sup> Reunión, la Secretaría siguió deliberando con los organismos y con los representantes del sector durante una misión a la China. El concepto de la ponencia de la Secretaría se repitió una y otra vez. Un organismo concreto se refirió a la posibilidad de mejorar la eficacia en el consumo energético de los acondicionadores de aire rediseñando y optimizando los intercambiadores de calor. Los representantes del sector se remitieron a los ejemplos de otras empresas homólogas en las que se han efectuado las conversiones conexas, específicamente las que pasaron a utilizar HFC-410A, y en las que los intercambiadores de calor sí han sido también modificados, y consideraron que la conversión de la

fabricación de los intercambiadores de calor sí estaba orgánicamente vinculada a la conversión de la cadena de producción.

12. En las deliberaciones mantenidas, la Secretaría se remitió, concretamente, a la decisión 61/44, del Comité Ejecutivo, por la que a la Secretaría se le pidió mantener la práctica vigente a la hora de evaluar las mejoras de componentes de los proyectos de conversión a HCFC para los sectores de equipos de refrigeración y de aire acondicionado, de manera que, tras la conversión, las características por las que se definen tales componentes permanecieran inalteradas o, de no disponerse de un componente dado similar, el componente en cuestión sólo se mejoraría en la medida necesaria para hacer posible la conversión. La Secretaría no cuestionó si la conversión de una cadena de producción de intercambiadores de calor estaría o no orgánicamente vinculada a la conversión del refrigerante, sino que, dado que técnicamente era algo innecesario y habida cuenta de las ventajas adicionales de explotación, la conversión tendría que ser sufragada por el Fondo Multilateral. Si bien la Secretaría no cuestiona el contenido técnico de los argumentos facilitados por uno de los organismos de ejecución al respecto de la eficacia energética, las actividades cuyo único y exclusivo fin es mejorar la eficacia del consumo energético no son admisibles para el Fondo Multilateral. En el caso en el que un beneficiario desee convertir la fabricación para mejorar el consumo energético, ello requeriría una financiación de contraparte o una financiación conjunta.

13. Se sugirió a todas las partes interesadas que debatieron con la Secretaría al respecto de este aspecto que presentaran por escrito todas las razones técnicas que justificaran el cambio del diámetro externo del tubo, pero excluyendo aquéllas conexas a la mejora de los costos y de la eficacia del consumo energético. Pese a haber establecido fechas límite de entrega, en las que no cabía confusión posible, y, en un caso concreto, de haber enviado un recordatorio al respecto, la Secretaría no ha recibido ni una sola contribución sobre el tema.

#### Recomendaciones de la Secretaría

14. Basándose en las deliberaciones indicadas *supra*, y en la decisión 61/45, la Secretaría sugiere al Comité Ejecutivo seguir las mismas recomendaciones que se recogen en el documento UNEP/OzL.Pro/ExCom/61/51 que se presentó a la 61ª Reunión, a saber: que el Comité Ejecutivo puede estimar oportuno, no considerar como costo adicional de explotación, en lo tocante a la conversión de sistemas de refrigeración o de aire acondicionado de HCFC a HFC no inflamables, los costos de capital conexas al reequipamiento de herramientas destinado a cambiar el diámetro de los tubos de los termointercambiadores aleteados, dado que esto se considera como una mejora técnica prescindible.

**ATTACHMENT I**

**A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR CONVERSION  
FROM R-22 TO R-410A**

# **A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR CONVERSION FROM R-22 TO R-410A**

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**Dennis Dorman**

**10/14/2011**

# **A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR CONVERSION FROM R-22 TO R-410A**

## **Summary**

The Multilateral Fund for the Implementation of the Montreal Protocol (MLF) supports the conversion of manufacturing facilities in developing countries that use ozone-depleting substances. A study was performed to determine the potential manufacturing cost changes for conversion from refrigerant R-22 to R-410A for fin and tube coils used in residential and unitary air conditioning products. Fin and tube coils are the dominant type of air-to-refrigerant heat exchanger used for evaporators and condensers of air conditioning equipment. R-22 was the refrigerant of choice for residential and small unitary air conditioning systems before its phase out in developed countries in 2010. R-410a is the current refrigerant of choice therefore a good candidate for conversion of products manufactured in developing countries. Since R-410A has different thermodynamic and physical properties, one might expect significant changes in heat transfer performance and the ability to retain pressure, especially in hot climates. It follows that coil design changes would be necessary and be accompanied by changes in manufacturing processes, capital equipment and tooling. However, the experience of major manufacturers in the period leading up to the 2010 phase out of R-22 is quite the opposite. The evaporator and condenser coils change very little and then only in regards to smaller features necessary to accommodate fatigue strength considerations. Therefore, major capital equipment purchase should be unnecessary. Some tooling cost may be necessary to manufacture the feature changes resulting from fatigue considerations while insuring robust manufacturing processes, but this should be small. Of special note is the need for manufacturers of coils to be able to rapidly run fatigue tests on coil features. A fatigue test facility is a non-trivial capital expense that must be borne by the manufacturer, or provided by an institution or private service provider that could provide the fatigue test service economically to several manufacturers on demand.

## **1. Introduction**

The Multilateral Fund for the Implementation of the Montreal Protocol (MLF) supports the conversion of manufacturing facilities in developing countries that use ozone-depleting substances. The MLF has asked for a study of the manufacturing cost changes that are necessary to change from R-22 to R-410A for fin and tube coils that are used in smaller air conditioning products, namely 2 to 5 kW residential room air conditioners (mini-splits) and 30 to 1000 kW unitary products. In order to determine the design changes associated with a refrigerant conversion, the refrigerant properties differences must first be considered.

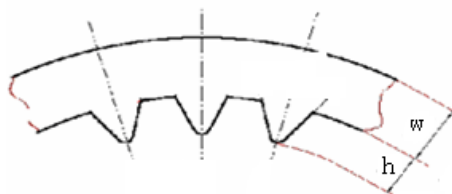
Since R-22 and R-410A have significantly different thermodynamic and physical properties, one might expect significant changes in coil design. It follows that manufacturing processes, capital equipment and tooling may need to be replaced or changed. This study will examine the two basic aspects of the design

problem, namely heat transfer and the ability to retain pressure especially under the extreme conditions that are prevalent in hot, arid climates. Manufacturing changes will be noted and their costs estimated.

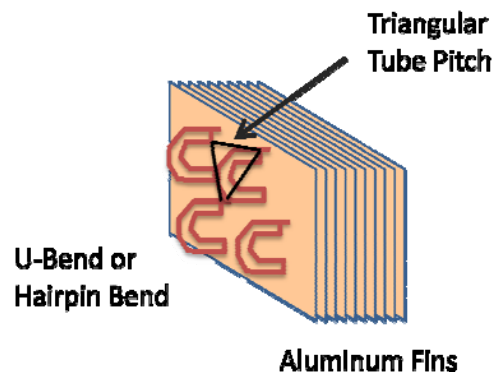
## 2. Baseline Coil Information

The following represent the typical base line coils or **design standard** that were used in products before conversion to R-410A. The term “**design standard**” means that most manufacturers would use coil design that are very similar though not identical. For example, the actual configuration of the aluminum fin may be different between manufacturers. The standards described below have been the used for more than 20 years. Enhancements to standard design occur as engineering and manufacturing technologies improve. Enhancements are usually aimed at improving performance and decreasing size and cost. These design standards are key to hitting ever increasing energy standards, maintaining a physically small product, and removing material cost – in other words, being competitive in the market place. The design standard is supported by raw material suppliers of fin stock, coil tubing, U-bends, etc., and are widely available from multiple suppliers. Companies that currently do not follow these design standards will be uncompetitive. Coil designs in developing countries may or may not be similar those described notably in the tube diameter which essentially changes all manufacturing tooling. In low volumes, the investment to upgrade typically is a barrier. These manufacturers would require a substantially larger upgrade of their manufacturing facilities than described in this report, leading to substantially higher conversion cost. While this would result in the ability to use R-410A and have competitive costs, it also represents an opportunity to increase product quality. Further discussion of such an upgrade is beyond the scope of this report

The term “**internally enhanced” tube (IE)** is used. This means that the tube has a raised surface on the internal surface similar to that shown below. The tube wall thickness noted in the detailed descriptions is the root diameter of the un-finned surface, “ $w$ ”. IE tubes have been available since the early 1990’s and the reason to change from smooth bore to an IE surface is purely economic, and has nothing to do with conversion to R-410a. The product design engineer has a fixed amount of cabinet space into which the coils must fit and changing the physical size of a cabinet is undesirable. In many cases, substitution of IE tubes instead of smooth bore would allow a change in the number of rows (from 4 to 3 in a condenser coil, for example) within the same cabinet size. A cost reduction would result, with no loss of performance. The design standard uses IE tubes as this type of tube has been widely available for many years. As product design engineers move towards higher efficiency within minimum space requirements, IE tubes are a practical necessity.



**2.1 Unitary (30 – 1000 kW):** The evaporator and condenser coils are contained within a complete packaged product, generally located on a roof. Equipment of this size and type generally has two or more separate refrigeration circuits each driven by one or more scroll compressors. The equipment responds to changes in ambient temperature or reduced cooling loads by turning a compressor off (called “unloading”) is achieved by turning off compressors within the circuit or an entire circuit. A coil slab is the complete manufactured coil. The coils may contain both refrigerant circuits in a single coil slab with the circuits intertwined to improve part-load performance. Multiple coil slabs are typically used for the higher capacity equipment



**Evaporator:** 3/8 inch O.D. IE copper round tube coils with configured aluminum plate fins mechanically bonded to tubes was the design standard prior to R-410A conversion. Many, but not all, coil designs had been changed to IE tubes prior to conversion. Again, use of IE tubes is not required for conversion to R-410A. The coils have 2 to 4 rows of tubes typically on a 1.0 inch triangular pitch. Coil height, length, rows, and number of refrigerant paths (coil tube circuits) varied by the refrigeration circuit capacity. A combination of hairpin bends and U-bends were used to connect tubes in each coil tube circuit. The U-bend to tube joints were flared and brazed. Tube wall thickness was .0118” to .014”. The first tube in each coil tube circuit was fed by a dedicated distributor tube connected to the coil tubes using either a crimped or flared brazed joint design. Each distributor tube was fed from a multiport distributor device. To maximize performance of the heat exchanger and minimize tube wall thickness required, the un-fined length of each tube protruding from the aluminum fins was kept to a minimum, usually around .5”. Overall refrigerant flow was controlled by a thermal expansion valve (TXV). Outlet headers, a larger diameter copper tube or pipe used to collect refrigerant from multiple tubes, use pierced or pierced and flared braze joints with mitered or saddle type joints for the gas outlet line.

**Condenser:** 3/8 inch O.D. IE copper round tube coils with configured aluminum plate fins mechanically bonded to tubes was the design standard prior to R-410A conversion. Many, but not all coil designs, had been changed from smooth bore to IE prior to conversion. Again, use of IE tubes is not required for conversion to R-410A. The coils had 1 to 3 rows of tubes on a 1.2 inch triangular pitch. Coil height, length, rows, and number of refrigerant paths (coil tube circuits) varied by the refrigeration circuit capacity. A combination of hairpin bends and U-bends were used to connect tubes in each coil tube circuit. The U-bend to tube joints were flared and brazed. Tube wall thickness was .0118” to .014”.



The first tube in each coil tube circuit was fed from a cylindrical header, and the last tube in each coil circuit was connected to a cylindrical outlet header all made from copper. The diameter of these headers varied by overall refrigeration circuit capacity with the largest outside diameter about 1.625" . To maximize performance of the heat exchanger and minimize tube wall thickness required, the un-fined length of each tube protruding from the aluminum fins was kept to a minimum, usually around .5". Both inlet and outlet headers use pierced or pierced and flared braze joints with mitered or saddle type joints for the gas inlet and liquid outlet lines

**2.2 Residential (2-5 kW):** Equipment of this size and type generally has a single refrigeration circuit driven by one non-unloading (no capacity control) rotary compressor. The evaporator coil is contained with a wall mounted cassette mounted in the air conditioned space, while the condenser coil is located outdoors.

Both the evaporator and condenser coils use 7 mm. O.D. IE copper round tube coils . Fins are highly configured aluminum plate fins mechanically bonded to tubes. The wall thickness is 0.25 mm. Though not the design standard, some manufacturers may have already switched to lower cost aluminum tube or smaller diameter (5 mm. or 6 mm.) before R-410A conversion. Small product typically has single row coils, whereas larger product uses two row coils. Both evaporator and condenser coils are made using hairpin bend and brazed U bends. In the case of evaporators, a short orifice is used to feed the circuits. Headers are made from small diameter copper tube. All connections are brazed.

**2.3 Manufacturing Processes:** The following describes some characteristic manufacturing features of the standard manufacturing process that were used for HCFC-22 coil manufacturing. Without going into detail, the standard manufacturing processes were generally used for both residential and unitary coils. The degree of automation, and the use of alternative processes is driven by volume/cost, manufacturing cycle time, manufacturing capacity investment considerations, and manufacturing quality control. By inspection it can be seen that most **tooling** would be sensitive to either O.D. or I.D changes (e.g. a fin die or hairpin bender), whereas the **major capital equipment** would not need to change (e.g. the fin press or bending mandrels)

**Fins:** Punched using a high speed progressive die fin press with automated feedstock and fin stacking. Fins are highly configured with features to improve air side heat transfer.

**Tube Cut Off:** Automated feed and tube length cut-off machine

**Hairpin Bender:** Hairpin bender with mandrels and automated feed

**Headers:** Punched or drilled with automated or semi-automated machines. T-drill or similar may be used for more robust brazed joint, especially in larger diameter headers

**U-Bends:** Purchased component (brazing filler metal rings may be included)

**Coil Structural End Plates and Supports:** Sheet metal is punched using a progressive pierce and bulge dies on a conventional punch press, or alternatively, an NC punch press with appropriate tooling

**Coil Assembly:** Manual or semi-automatic

**Tube Expansion:** Ball end multi-rod expanders or ball expanders driven with hydraulics for low volume coils

**Headers, U-bend or Distributor Tubes:** The coil tube is crimped to form a brazing joint for the distributor tube. Brazing is done with a dry nitrogen purge. Single and multi-tip torches are used.

**Pressure and Leak Test:** Air under Water immersion in water tank with safety cover

**Final Product Pressure and Leak Test:** Dry air plus halogen leak detector

**Process Fluids:** All process fluids used during manufacture and testing of the heat exchangers are selected to be compatible with R-22 and the refrigeration mineral oil

### 3. Heat Transfer and Refrigeration Cycle Performance

The calculated coefficient of performance, COP, for R-410A vs. R-22 in a refrigeration cycle at a variety of conditions shows that R-410A should be 5% to 7% lower than for R-22. The efficiency of a refrigeration cycle is typically provided as the COP, and is calculated by using the refrigerant thermodynamic properties, as the cooling effect divided by the energy, i.e. the electricity needed to drive the cycle. The experience of major manufacturers has shown that this is not the case. When R-22 is replaced by R-410A in a refrigeration system the performance is nearly identical. Some studies have actually shown a slight improvement with R-410A. This anomaly is explained by the superior transport properties of R-410A which result in improved heat transfer and lower component pressure drop in the evaporator and condenser. This improved heat transfer performance overcomes the poorer cycle COP resulting in nearly identical overall equipment performance. Product rating programs used to report performance of equipment provide for variation in component performance and manufacturing by allowing tolerances, typically 5% or more. The differences in coil performance between R-22 and R-410A were small enough to not be reflected in catalog or electronic selection programs for the equipment, once compressor flow adjustments were made. Therefore, coils were not changed. In other words, changes to heat exchanger face area, fin spacing, number of rows, and number of coil circuits are not needed when changing from R-22 to R-410A.

However, the critical temperature for R-410A is 43 degrees F lower than for R-22. The closer a refrigeration cycle operates to the critical temperature for the refrigerant the lower the cycle COP will be. Studies have shown that R-410A performance, both capacity and COP, decreases more rapidly than R-22 as the ambient temperature and the cycle condensing temperature is increased. This effect begins to be significant (>5%) at ambient temperatures above 50 C. The practical importance of this effect is minimal. Residential and unitary systems are typically designed for 40 C. At elevated temperatures, a fall off in performance occurs, as well as a fall off in refrigeration capacity. Usually refrigeration system capacity is oversized to insure that comfort is not lost and the fall off in performance is accepted. It should be noted that operation at 45C to 50 C is an extremely unusual application, lasting for relatively short periods of time annually, though they can last for a large part of successive days. For example, weather data for Abu Dhabi located in the United Arab Emirates, Middle East Region, shows that only 0.4% of the time does the ambient temperature exceed 45 C. So, manufacturers do not typically change coils for the purpose of providing product to hotter climates. This remains true for R-410A.

## 4. Pressure

The pressure of R-410A compared to R-22 is nominally 60% to 70% higher at a given temperature, a very large difference. The subject of coil changes that are necessary to withstand the pressure difference revolves around the design practices and governing codes and standards. Coils do not fall under the rules of pressure vessel codes, such as ASME Section VIII or similar pressure vessel codes. Coil design will normally fall within local codes or standards.

One widely used standard, ASHRAE 15, **Safety Standard for Refrigeration Systems**, and its companion standard, UL 1995, **Heating and Cooling Equipment**, govern product safety for end use air conditioning products in North America. ASHRAE 15 has its origins in the U.S. but it is a globally recognized and adopted standard. A similar but not identical standard, ISO 5149, is used in Europe. UL standards are frequently adopted international as well. These standards and others like them in various countries provide the basis for pressure design of coils and other components such as compressor that do not fall under the pressure vessel codes.

Using UL 1995, one manufacturer reports that the design pressure for the R-22 coils was set at 450 psig, corresponding to 160 F. In this case, the design pressure is not set by the ambient temperature for the application, but rather by consideration of temperatures that may be experienced during shipment, usually in a non-ventilated closed truck or trailer parked in a hot location such as Arizona. This means that application in hot ambient temperatures is already within the design pressure. Per UL 1995 (Clause 61) the design is required to pass a burst test with a minimum burst pressure of 2250 psig or 5 times design pressure. For R-22, the coils achieved 2250 psig using the standard design and standard manufacturing methods. Employing the same test method and strength requirements to coils using R-410A would result in a minimum burst test pressure of 3900 psig. Designing coils to meet this pressure would not be practical, since the tube wall thickness increases proportionately with pressure. Therefore the wall thickness would have to be 70% higher ! The additional copper would be very costly. Also, the flow area within the tube is decreasing as the square of the diameter change, and the pressure drops would increase accordingly. Even though pressure drop a pressure drop change is naturally offset by the lower volumetric flow rate per kW cooling (30% lower for R-410A) and re-circuiting the coil can also provide relief, we would still expect that the increased parasitic loss would be a significant disadvantage. So an engineering solution must be used.

The UL burst test requirements using a design margin of 5 times design pressure is a simple method of demonstrating adequate margin of safety against pressure. National codes and standards typically allow a more careful examination of the actual conditions through more sophisticated tests and/or analysis. See Section A1 for more discussion. Regarding coils used in air conditioning products, UL 1995 Clause 62 (Fatigue Test Analysis), provides an alternative method. This method is a more sophisticated alternative engineering method, employs well known and demonstrated engineering principles, and is very rigorous because it subjects samples to real world stresses. For this method, test samples are subjected to a 250,000 cycle pressure test between low and high side design pressures for the actual application, followed by a burst test at 3 times the design pressure. This method recognizes

the real application, where pressure changes occur during cooling cycles that are the result from shutdown when the coil cools to ambient to a higher pressure from operation. Also, a burst test pressure is required at 1950 psig or 3 times design of 650 psig. The test must be repeated annually on all discrete heat exchanger designs in a manufacturer's product line, essentially a quality control function on design change control and manufacturing processes. Manufacturers of R-410A equipment have found that most existing coil heat exchangers designed for R-22 pass this test with minimal design changes, but with some feature changes, such as the length of un-finned tube protruding from the coil face and end support. Universally they report that there are no coil failures with either R-22 or after conversion to R-410a.

**The method described above is equally applicable to smooth bore tubes and IE tube coils and any tube diameter 3/8 inch and below. See Section 5 for other diameter tubes, not meeting the design standard of Section 2.**

Manufacturers report many fatigue test failures during qualification testing. These are areas of weakness. They are discovered and resolved, one by one. Design feature changes, and the use of higher quality components generally will solve the design problems that are found. Areas of particular importance are:

- Coil heat exchanger tubes must be free of defects such as dents and scratches. Damaged tubes will always produce a fatigue failure.
- The length of coil tubes not covered by fins must be kept to a minimum. Recall that in the R-22 designs, 0.5 inches of length was common. This was reduced for R-410A designs. This is particularly true for the heat affected zone in tubes outside of the coil casing that are brazed to U-bends or header stubs. The fins provide support for the tube and increase the burst strength of the tube
- Header joint designs need to include reinforcement such as saddle type or flared holes that provide sufficient overlap of material for a sound braze joint. This means that a T-drill or similar is necessary. Cost of T-drill tooling is typically less than \$5000 per drill head.
- U-bends are generally purchased and it may be necessary to increase the wall thickness of these parts since they will thin during U-bend manufacture.
- The crimp joints that are sometimes used for distributor to coil attachment will not always be sufficiently strong. Designs may need to be changed to a flared end distributor or a purchased flared adapter for this joint. Tooling changes for these features are typically less than \$10,000 per station.
- For headers larger than 1.375" diameter "K" wall thicknesses will probably be required (Reference ASTM B88. This means that wall thicknesses for the particular diameter used is about 20% increased). The heavier wall tube should work on up to 1.625" diameter headers. This does not eliminate the need for high quality saddle or flared header to tube joints.

- Brazing quality must be carefully controlled. Especially important are standard brazing procedures and qualification of the manufacturing technician, use of a nitrogen purge during brazing and routine inspection to insure quality. Nitrogen purge is used to insure cleanliness inside the refrigeration system and pressure integrity of the joint. Nitrogen is typically provided in bottle form at the brazing station. Bulk tanks and piping could be used for very high volume production. Careless brazing is the largest single source of leaks, which is the largest single warranty expense for manufacturers, and is especially problematic with higher pressure R-410A.

Purchase or lease of fatigue test equipment that can induce rapid pressure cycles using hydraulic fluids will be a direct cost associated with the changeover to R-410A. This cost will vary depending on the size and number of testers required to support a particular facility. One manufacturer of unitary equipment reports a capital cost of \$100 K for a fatigue test facility for a multi-port tester, so the capital cost is not trivial. This facility tests multiple large coils at once, and was booked solid during the years of engineering work leading to R-410A conversion. A single port tester designed for smaller residential coils is estimated to cost \$30,000. All facilities are automated to rapidly run the fatigue test in a reasonable period of time. Facility cost is more a function of the number of ports than the physical size of the tester.

An institution or private service provider could purchase and install the necessary facilities for use for a group of manufacturers. In this case the service is provided as an expense, rather than a capital acquisition or lease. One manufacturer reports that the cost of a single test at an agency is approximately \$4000. Multiple tests of a typical design would be normal to pass the fatigue requirements. Then the design features would be used over an entire product family. However, since coil features do change over a product line (additional rows and differences in headers for example), it would be prudent that the final designs of each coil be final tested wherever any design feature is different. In doing so it would not be unusual to have a finding or two. The total expense can be estimated and compared with investing in a test facility. For purposes of the exercise assume 5 sizes of product within a family are manufactured, each with slightly different features that affect pressure integrity. Further assume that a small single port tester costing \$30,000 could serve the need for qualification testing.

Basic design qualification = 4 tests @ \$4000 = \$16,000

Final check of each size, with 2 retests due to test findings = (5 + 2) @ \$4000 = \$28,000

Total Expense = \$44,000

Cost of a Test Facility = \$30,000 + labor to run the tests + incidentals (energy, spare parts, maintenance)

The above represents a realistic number of tests for the given scenario. Obviously the number of coil variations is driving a large part of the expense. In this case, depending on financing available to a particular company, a rational choice can be made between expense or capital investment. Since there are ongoing expenses to run a test facility, the capital investment choice is typically more attractive if

there is already a permanent lab facility that is engaged in various testing activities as product designs continue to evolve.

As previously noted in the R-22 baseline discussion, process fluids and mineral oils are carefully chosen and qualified to insure compatibility with R-22. Systems using R-410A employ synthetic POE oils, or PVE oils, so different process fluids that do not cause chemical interaction must be used. This is essential engineering work and failure to pay attention to these details will result in breakdown of the oil, and ultimately lead to very expensive compressor failures.

Most manufacturers using round tube heat exchangers for R-410A continue to use 3/8 inch tubes for all circuit sizes. The lower volume flow rates of R-410A allow coils tubes to drop to 5/16" diameter (unitary) or 5 mm. or 6 mm. tubes (residential). Residential product will likely see more pressure to use all aluminum coils. All of these are driven by lower cost and will happen over time. These are not related to a R-22 to R410a conversion per se, although the high density R-410a makes a change to smaller diameter tube attractive, since pressure drop, a parasitic loss, will not be excessive.

Finally there needs to be a remark about maintaining performance. Maintaining performance during an R-410A conversion is non-trivial manufacturing engineering and quality control work. Some areas to pay close attention are:

- **Coil Circuiting:** Since the volumetric flow rates (or CFM/ton) are different, it follows that coil circuiting will likely change such that the tube surfaces are fed as nearly uniformly as possible. In doing so, performance is maintained or in some case increased. Coil circuit changes generally do not involve tooling changes, just assembly change.
- **State of Coil Tooling:** The tooling used in manufacturing simply wears out or need maintenance from time to time. A good example is a fin die. Worn dies lead to cracked collars, the main bonding area between the tube and the fin. A poor bond due to a cracked collar will degrade coil performance significantly. No conversion program should be undertaken without an examination of the state of the tooling.
- **Brazing:** It seems that the universal experience among manufacturers is the inability to maintain high brazing quality. Leaks and other failures within the coil or product assembly, usually found in the field, are very expensive. The higher pressures of R-410A make this problem especially noteworthy.
- **Coil damage during assembly, test, packaging and shipment:** Experiences is that R-410A coils are much more sensitive to leak or failure than R-22 as a result of manufacturing or shipping damage to coils.

## 5. Coil Designs not meeting the Design Standard

As discussed in Section 2, prevailing design standards and the use of the methods describe in Section 4 yield a successful conversion from R-22 to R-410A. **Essentially, any tube diameter 3/8 inch and below can be successfully converted without major cost penalty or investment.** It is logical to ask, what happens if the design does not meet the design standard, notably in the area of tube diameter?

Reaching back in time, coils were made with 5/8 inch (usually used with R-12, a lower pressure refrigerant) and ½ inch tubes. So it is possible that an older legacy design is currently manufactured for R-22 coils.

As discussed earlier, the tube wall thickness increases linearly with pressure. It also increases linearly with tube diameter. So a ½ inch tube, for example, would have a tube wall thickness two times that needed for R-22 (70% for pressure, 30% for diameter). This is twice the amount of copper and an enormous cost penalty. So, increasing the tube wall thickness, either on the I.D. or O.D. would not be an attractive solution. Assuming that the alternative fatigue method could be employed is a reasonable assumption, but there is really no experience among major manufacturers that would validate it. One other potential solution is a change in the tube alloy for increase strength. But then, special alloys are not cost free, since there is little supply base for this type of coil tube. One must assume that the solution is some combination of the potential changes – thicker walls, different alloys, and use of the fatigue method – but the exact solution will depend on the singular case.

## **6. Summary of Manufacturing Costs for Conversion**

In Section 2, a summary was given of the primary manufacturing processes and tooling. Given below is a summary of the manufacturing tooling changes that might be expected for a typical conversion. There is no major capital cost for conversion.

**Fins:** NO TOOLING COST

**Tube Cut Off:** NO TOOLING COST

**Hairpin Bender:** NO TOOLING COST

**Headers:** T-drill or similar must be used. T-drill is a trade name and other alternatives exist to raise a more robust brazing collar on the header. T-drill heads cost approximately \$5000 per drill head and can be used with the either manual or automated drilling equipment. The basic drilling equipment itself does not change. The number of drilling machines and type is widely variable depending on production volumes.

**U-Bends:** NO TOOLING COST

**Coil Structural End Plates and Supports:** NO TOOLING COST

**Coil Assembly:** NO TOOLING COST

**Tube Expansion:** NO TOOLING COST

**Headers, U-bend or Distributor Tubes:** Crimping of distributor tube to coil tube estimated to cost less than \$10,000 per station. A station is the production facility where the work is performed, usually one per production line.

**Pressure and Leak Test:** NO TOOLING COST

**Final Product Pressure and Leak Test:** NO TOOLING COST

**Process Fluids:** Process fluids must be carefully examined for compatibility with R-410A and POE oil. It should be noted that PVE oil is also used with R-410A. This is engineering work, but not a manufacturing cost.

Though not a manufacturing tooling cost, a **Fatigue Test Facility** cost is approximately \$100,000 for a large, multiport tester but could range down to \$30,000 for a small single port tester.

In cases where current coil designs at a manufacturer do not meet the design standard and it were possible to create an engineering solution as discussed in Section 5, the same capital cost requirements shown above may be assumed. The number of tests needed to qualify a larger diameter tube may be higher than that experienced on small diameter tubes (3/8 inch and below). Also, such manufacturers are also likely to incur significantly increased per-unit costs through the increased use of copper and/or more pressure resistant alloys, whichever may be applicable.

## 7. Conclusion

The work done in the period leading up to the 2010 phase out of R-22 by major manufacturers indicates that R-410A coils do not require major design changes. Essentially an R-22 design standard coil will yield the same or better heat transfer and handle the higher pressures. Neither the tube O.D. the I.D, nor the standard materials of construction needed to change. Coil circuiting changes are common to optimize performance within the design space. Regarding pressure, including but not limited to the higher temperatures of the places like the Middle East, coils cannot withstand the typical burst test pressures used to qualify R-22 coils, and an alternative method to insure safety must be used for R-410A. Engineering work must focus on fatigue strength rather than simple burst test limits. The broad experience is that a good analytical and test program likely produce a successful design without major change, but small feature changes (such as unsupported tube length). Therefore, the capital equipment to manufacturer R-410A coils is essentially unchanged. Small tooling changes may very well be necessary, but these are low in cost. For example, a T-Drill head used on a manual or NC drill press is less than \$5000. Of special note is the need for manufacturers to be able to rapidly run fatigue tests on coil features. A common experience was to try a variety of design features in areas of fatigue failure. A facility large enough to handle the largest unitary coils cost approximately \$100,000. A fatigue test facility is a non-trivial capital cost that must be borne by the manufacturer, or provided by an institution or private company that could provide the service economically to several manufacturers on demand. Finally, attention to details, such as process fluids changes, the state of manufacturing tooling, and quality brazing are among the items that cannot be overlooked for a successful conversion. This is engineering work, not generally related to capital or tooling costs.



## Annex A

Annex A is reserved for more detail on certain subjects or to answer several additional questions that were posed in the inquiry that have not been addressed, but are not central to the issue of manufacturing conversion costs when changing to R410A.

### A1. Pressure Design

The subject of pressure design is somewhat complex. Two basic philosophies exist within the global community of codes and standards.

1. Simplified design by rules
2. Sophisticated design by analysis / test

For example, ASME Pressure Vessel Code, Section VIII Div. 1 for the design of unfired pressure vessels gives a thorough, albeit simple and conservative, set of design rules for safe design of pressure vessels. Alternatively, for unusual designs or in cases where there is unusual manufacturing or costs implications a alternative method is given in Div 2. Both methods produce safe design for pressure containment but the Div. 2 is much more sophisticated and demanding.

Pressure vessel codes do not exist for coil designs and other components such as compressors. In these cases, other standards can be found within the international community. For example, in North America, ASHRAE and UL standards are used. In Europe, ISO 5149 and IEC PED are parallel standards that are used. For automotive applications, SAE J 6.9 is widely used throughout the world. All such standards follow a similar pattern to that describe above. These standards usually allow a simple method, or alternative methods using more sophisticated tests and/or analysis.

Regarding fatigue testing, well know engineering methods are applied, but also proven before the standard is adopted. In the case of UL 5149, in the mid-1990's work was done within UL and with several manufacturers to demonstrate safety and equivalency of the alternative methods. In doing so, a realistic assessment of the **real loading and stresses** is necessary. UL reports that over 300 independent tests were done by them, independent of hundred of tests done by various manufacturers to demonstrate that the alternative method that UL finally adopted was indeed safe and equivalent or superior to a simple one- time pressure test. ISO 5149 contains a similar method.

### A2. Alternative Refrigerants

While this study is not intended to discuss alternative refrigerant choices in great detail, discussion of conversion to a number of alternative refrigerants was requested. R-410A is the current refrigerant of choice for developed countries for unitary and residential air conditioning systems and therefore a good choice for developing countries. However, any refrigerant choice is ultimately a trade-off between ODP, GWP, safety and performance as well as wide commercial availability for service. See table below for a comparison of select refrigerants that are suitable for smaller residential and unitary air

conditioning systems. Items highlighted in *Italics ==* below indicate a significant shortcoming. Items highlighted in underlined *Italics ++* are best in class.

### Comparison of Refrigerant Properties for Select Refrigerants

|        | ODP            | GWP            | Simple Cycle COP*      | ASHRAE Class |
|--------|----------------|----------------|------------------------|--------------|
| R-22   | <i>.034 ==</i> | <b>1700</b>    | <i>6.98 ++</i>         | <b>1</b>     |
| R-410A | <u>0</u>       | <b>1725</b>    | <b>6.485 (-7%)</b>     | <b>1</b>     |
| R-32   | <u>0</u>       | <i>650 ++</i>  | <b>6.735 (-3.5%)</b>   | <b>2L ==</b> |
| R-407C | <u>0</u>       | <b>1610</b>    | <b>6.744 (-3.4%)</b>   | <b>1</b>     |
| R-404A | <u>0</u>       | <i>3300 ==</i> | <i>6.282 (-10%) ==</i> | <b>1</b>     |

*\*The simple cycle COP does not reflect any positive transport properties benefit discussed in Section 3.*

As can be seen in the table, there is no perfect refrigerant – all have some shortcoming. Among the refrigerants suitable for application for comfort cooling in residential and unitary equipment (i.e., a low CFM/ton), R-410A has zero ODP and is an ASHRAE Class 1 refrigerant (means no unusual safety restrictions). It is a binary blend of 50% of R-32 and 50% of R-125 and accordingly has a relatively high GWP (1725). This means that it will likely come under regulatory pressure in time future. A smaller number of systems, notably in Japan, use R-32 today. R-32 has zero ODP and a low GWP (650), but it is a moderately flammable that was recently reclassified from ASHRAE Class 2 to 2L. Japanese codes have been relaxed to allow its use, but this is not the case in many countries, especially so in North America. Class 2L refrigerants are unusual in that they are characterized by very low flame speeds leading to weak and unstable ignition events, and are difficult to ignite due to their high Lower Flammability Limits (LFL). Therefore, there is a technical basis for relaxing application codes for Class 2L refrigerants. ASHRAE 15, **Safety Standard for Refrigeration Systems**, is in the process of introducing special rules to allow the use of Class 2L refrigerants. The rules are expected to be only moderately more restrictive than Class 1 (non-flammable), making their possible use more popular. R-407C, a ternary blend of 23% of R32, 25% of R125 and 52% of R134a, was used for a short period of time notably in Europe and to a lesser extent in North America. It has zero ODP and a high GWP (1610). Unlike other blends and single component refrigerants, it fractionates within a heat exchanger and produces a non-constant condensing or evaporating temperature (known as ‘glide’). This causes higher compressor lift and higher power, not reflected in the calculated simple cycle COP value. It is not widely used and survives today as a zero ODP transitional refrigerant for legacy products that are not economical to redesign in a new refrigerant. R-404a was also used briefly, notably in Europe. It is a very poor performing refrigerant in air conditioning systems, and it is not used today in any major product line. Any of these refrigerants can likely be substituted into a R-22 coil design and produce an acceptable design without major change, essentially for the same reasons that R-410A can. At this juncture, the refrigerant producers have not yet suggested a clear alternative to replace R-410A. Again, finding a refrigerant suitable for small air conditioning systems with zero ODP, low GWP, high performance and no unusual safety code restriction is no easy task.

Conversion to R-410A in many developed countries was accompanied by performance increases that are required by governments or national codes or standards. It is becoming more understood that the major component of global warming is the **power consumption** of the air conditioning product during its lifetime of use, **not the direct GWP effect of refrigerant leaks**. Refrigerant leaks are insignificant in smaller air conditioning systems but not without engineering effort and attention to quality details in manufacturing.

### **A3. Coil Efficiency**

Two inter-related questions concerning coil efficiency were posed in the inquiry.

1. What is the effect of an increase in the air side surface of 10%, 20% and 50%. This question is essentially the same as a second question in the inquiry relating to the effect of a change in evaporating and condensing temperature.
2. Comment as to the built-up of dust or dirt at the heat exchanger during regular operation, assuming rare cleaning, and how common these conditions are.

The power consumption of the end use product is the usual performance metric expressed as COP or EER at a set of standard temperatures, or Seasonal Energy Efficiency Rating (e.g. SEER) which represents a mix of temperatures encountered during a typical cooling season.

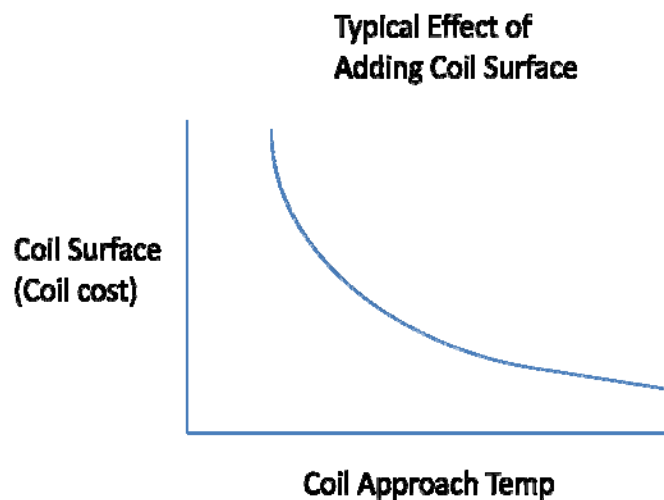
The difference between the saturated evaporating temperature and the saturated condensing temperature is the pressure difference or 'lift' that the compressor must work against. The smaller the difference is, the lower the power consumption.

The difference between the ambient temperature and the saturated evaporating or condensing temperature is known as the 'approach'. The approach temperature is one measure of coil performance. The lower the approach temperature, the lower the lift, and therefore the lower the total power consumed by the compressor (offset by any increase in fan power). A picture may help .

A variety of coil feature can be changed to improve heat transfer performance and lower the approach temperature in any coil. These include:

- The configuration of the aluminum fin surface (usually proprietary and patented features)
- Increased fin spacing (but fan power is negatively affected due to increased pressure drop)
- Refrigerant metering and distribution (uniform distribution is desired)
- Increased air flow across the coil (but higher fan power negatively affects total product power consumption)
- Enhancement of the internal surface of the tube
- Tube pitch (described in Section 2.1)
- Fluid flow within the circuit – inter-weaving of tubes
- Coil surface increase through additional number of rows or total surface area (but air flow and fan power must increase too)

The engineering of coils is complicated, since there are trade-offs. For example, increasing the fin spacing improves the approach temperature, but at the expense of additional fan power. So, overall the product performance may not improve at all. Certainly adding surface through surface area or additional coil rows will lower the approach temperature, but it is costly and decreasingly effective. Increasing coil surface follows the law of diminishing returns, until additional coil surface will have no positive effect at all as shown in the graph.



The approximate range of coil approach temperatures for evaporator and condenser coils is between 20 °F (1 row smooth bore tube coil) to 10 °F (4 row IE tube). 10 °F is nearly a practical limit.

## A4. Coil Fouling

The effect of coil fouling has been studied by various manufacturers. The studies are proprietary and the data cannot be disclosed. However, a couple of statements can be made.

- Actual fouling in the field is not determinate since there is a wide variety of conditions that can cause fouling (dust, moisture, debris from plants, etc). However, the nominal performance penalty is approximately 1.5% every year that the coil is not cleaned.
- Outdoor coils typically become more fouled and fouled more quickly, than indoor coils.
- The use of highly configured fins, notably with louvers or slit fins, accelerates fouling.
- Coils can be restored to their original as built performance using conventional cleaning methods describe by the manufacturer and/or provided by competent service providers. This typically involves use of a special, non-toxic, non-acid based cleaner that foams to remove dirt and grime and push it away from the inner part of the coil. In the case of the indoor coil, it is sprayed on but not rinsed (dehumidification will remove residual cleaner). In the case of the outdoor coil, the cleaner is sprayed on and washed off.

Worse than fouling, **coil corrosion** is a topic of great concern especially in coastal regions. Normal aluminum coils can be severely and rapidly affected by salt laden air in these regions. The main issue is that a salt laden air sets up an electrolytic cell between the copper tube and the aluminum fin. The more reactive aluminum fin corrodes rapidly, sometimes within a few months in severe cases. Performance is dramatically affected, especially when the corrosion occurs at the aluminum to copper interface. This may be estimated to be on the order of several percent annually. In severe cases after only a few years, a complete loss of the tube to fin bond causes the heat transfer mechanism to be lost, and the product will not operate properly due to high condensing temperature causing the product to trip its high pressure safety control. Various techniques are used to overcome the problem including dip and bake coatings and spray-on coating and pre-coated aluminum fins. The degree of effectiveness of these treatments is widely variable, and there is no standard solution within the industry that is proven and used by most manufacturers.

**ATTACHMENT II**

**VIEWS EXPRESSED RELATED TO RETOOLING FOR MANUFACTURING  
HEAT EXCHANGERS**

## Annex I

### **EXCERPT FROM THE CONTRIBUTION OF ONE DELEGATE TO THE DISCUSSION DURING THE 62ND MEETING OF THE EXECUTIVE COMMITTEE (TRANSCRIPT)**

... “We all are aware that alternatives to HCFC-22 like R410A or HFC-32 and R-407A are inferior in thermodynamic properties like HCFC-22. The TEAP report clearly brought out that the refrigerants like R-410A which is not only having the high pressure in the system because of its lower normal boiling point but also has lower critical temperature in comparison to HCFC-22. Similar will be the case with the HFC-32. The TEAP recent reports have also clearly brought out that the cooling capacity and the performance of the system with the refrigerants like R410A is much lower than that of HCFC-22 and it decreases very forced with the increase in condensing temperatures. The drop in capacity and the performance is as high as 15 to 20 per cent.

Mr. Chairman, as all of us are aware that most of the Article 5 Parties are in tropical climates where AC systems are designed and tested at the 55 degree centigrade condensing temperature or above. As such, condensing temperature, the cooling capacity and the performance of refrigerants like R410A or HFC-32 is quite lower than that of HCFC-22 maybe in the order of 15 to 20 per cent. The industry is also to provide to the consumer the same capacity units as they are supplying or marketing yearly there as they were supplying same performance if not better.

Mr. Chairman, we have been given to understand by your industry and the experts working in this AC systems, especially the air cool systems that the heat exchangers, especially the condenser plays an important role in the performance of these units working in tropical climates where the systems are already operating closer to the critical temperature. Any further increase in condensing temperature due to not redesigning the heat exchangers for the change in alternatives refrigerants or properly optimizing it will have an impact on the cooling capacity as well as the significant decrease in its performance. We appreciate the efforts of the MLF Secretariat to present a complex technical subject in a simple manner but the simplification must not unremain the realistic technical requirements of this competence especially the heat exchangers. The paragraphs 4, 5, and 6 of the MLF Secretariat document gives the process of manufacturing of the refrigerant to air heat exchangers which is very nicely explained; however, no attempt has been made to see that thermodynamic are heat transfer requirements of the replacement of refrigerants like R410A and maybe HFC-32 and their impact on the design of these heat exchangers. In our opinion the process of manufacturing follows the design of such competence which is based on the thermophysical properties and the flow characteristic of the refrigerant to deliver the required capacity and performance.

Mr. Chairman, we have been given to understand by our industry and the heat exchanger experts who are involved in developing in the manufacturing of such heat exchangers that the heat exchanger design is quite critical and it is complex and it cannot be simply referred as and dispensed as MLF Secretariat paper. I quote as it reads the last sentence of paragraph 6 of the Secretariat paper, document UNEP/OzL.Pro/ExCom/62/55: “These heat exchangers do not substantially differ in their design or material between HCFC-22 technology and the various current HCFC-22 alternatives (except for ammonia and CO<sub>2</sub>).” This type of statement could not be appreciated by the experts and as per experts there is a strong reservation about the technical expertise which this paper has been prepared.

The design of such heat exchangers is quite complex as there is two-phase flow of refrigerant, which is either condensing or boiling as the case may be. The inside heat transfer coefficient is a function of mass flux under volume flow rate, thermophysical properties of refrigerants as well as flow regimes which are set during the process of condensation. The material used include surface softness of the tube influences the heat transfer coefficient as well as pressure drops in the heat exchangers which have direct influence on the performance of the heat exchangers and the air-conditioning system. The reduction of the mass flow rate as well as the volume flow rate of the refrigerant as it is expected with R 410A or HFC-32 is likely to be reduced in the order of 25 or 30 per cent. This reduction is likely to change the flow regime

maybe from turbulent to laminar meaning drastically dropped inside the heat transfer coefficient if the diameter of the tubes is not changed accordingly.

Mr. Chairman, I would like to mention that presently most of the manufacturers of the heat exchangers are internally grooved of fin tubes instead of substitutes which further make the situation more complex. Mr. Chairman, we have been told by the experts that the heat exchanger design is so complex even same number of tubes of same diameter used but arranged in different manner like circuiting heat exchanged or arrangement of tubes from stagger to inline or inline to staggered, the performance of the heat exchangers drastically changes. The outside surfaces are fins.

These surfaces also play a vital role especially arriving on overall heat-transfer coefficient of these heat exchangers and finally for the size of the heat exchanger for a given capacity. The fins spacing or fin density also influence the overall heat transfer coefficient

However the increase in the number of fins vary with the length of the tube also influences hence decrease performance. Such a complex design cannot be said as what has been mentioned in the Secretariat's paper. I would like to raise the issue of need of using better quality copper tubes in the case of HFC refrigerants like R-410A or HFC-32 as referred by the MLF Secretariat paper. The combination of HCFC-22 and the minimal uses presently is very tolerant and very mild and allow the use of normal refrigerants, commercial copper tubing as this was also the case in the use CFCs. The industry in Article 5 Parties mostly uses such tubes. We recall the period when our industry changed from CFC-12 to HFC-134a, there were issues related to copper plating and especially on the release valves of the compressors this is mainly related to the quality of tubing used. In case of HFC-134a and the polysteryl combination because of aggressive properties, the change in quality of copper tubing was essential. Similar will be the case for R410-a and HFC 32, the quality of copper tubing is essential for the changeover.

Mr. Chairman, in summary, I would like to emphasize that redesign and retooling of heat exchangers is an integral part of the conversion from HCFC-22 to refrigerants like R-410A and HFC-32 and in no way this could be said can this be said avoidable upgrades thus associated cost or the IC or IO cost, these need to be considered in the project proposal. We propose that the decision on heat exchangers should address the following elements: 1) the cost of redesign and/or changes to heat exchangers processing required for addressing different properties of alternatives should be eligible in full (cleanliness levels for copper tubes under processing due to hydroscopic lubricants, ensuring heat tightness, etc. because they are directly related to phase-out of HFCs; 2) the cost of redesign and/or changes to heat exchangers required to maintain capacity and the performance of the heat exchangers at the baseline level as well as to maintain climate impact at the baseline level should be eligible in full. As required by MOP decision XIX/6, adverse impact on health and environment have to be minimized and the climate benefits maximized. The above should apply to both incremental capital cost and operating cost required for implementing the redesign and/or changes in the heat exchangers under processing. Thank you Mr. Chairman.”

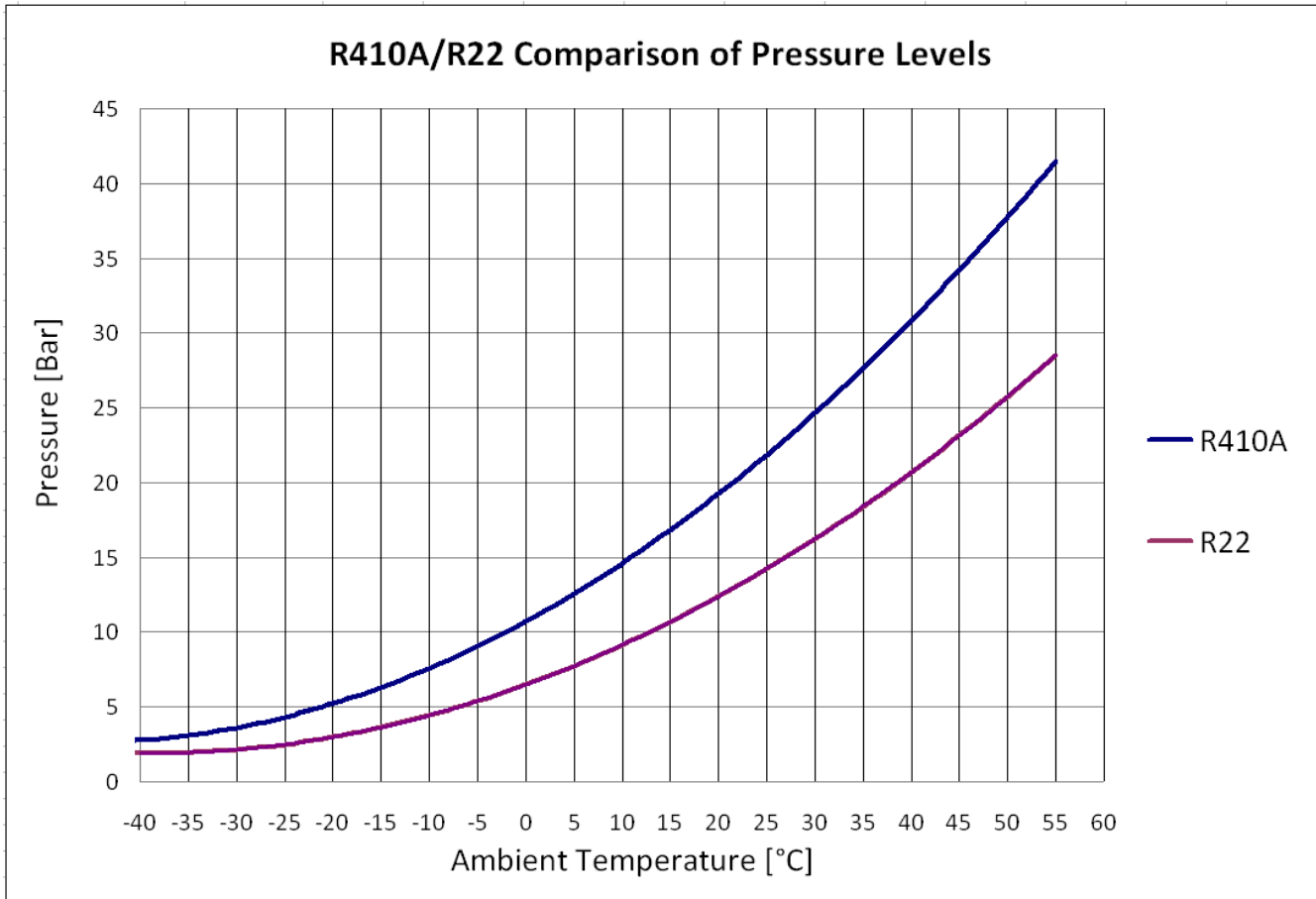


Annex II

**“ R410A/R22 COMPARISON OF PRESSURE LEVELS”, SUBMITTED BY PETRA  
ENGINEERING R&D LABS**

## R410A/R22 Comparison of Pressure levels

- R410A refrigerant is a near azeotropic blend. It is widely used in air conditioning applications.
- At high ambient temperatures, energy consumption/COP initially seems to be less favorable than with R22. This is mainly due to the thermodynamic properties of this refrigerant. On the other hand, very high isentropic efficiencies are achievable (with reciprocating and scroll compressor), whereby the differences are lower in reality.
- The material compatibility is comparable to the previously discussed blends and the same applies for the lubricants. However, the pressure levels and the higher specific loads on the system components need to be taken into account.
- The fundamental criteria for HFC blends also apply to the system technology with R410A, however the extreme high pressure levels have to be considered (42 bar at 55°C Ambient temperature).
- Compressors and other system components of “Standard Design” have substantial limitations for the application of this refrigerant. However, due to the favorable properties of R410A considerable effort is taken for the development of suitable products.
- When considering to cover usual R22 application ranges, the significant differences in the thermodynamic properties (e.g. pressure levels mass and volume flow, vapor density) must be evaluated.
- This also requires considerable constructional changes to compressors, heat exchangers, and controls, as well as measures of tuning vibrations.
- In addition, safety requirements are concerned also affecting the quality and dimensions of piping and flexible tube elements (for ambient temperatures of approx. 55°C/42 bar).
- Another criterion is the relatively low critical temperature of 73°C. Irrespective of the design of components on the high pressure side, the condensing temperature is thus limited.



## Clarification why changes in the production lines in PETRA company is required for changing from R22 to R410A and why to consider this incremental operating cost of the heat exchanger of PETRA project?

\*Changing the toolings and the heat exchanger production machines in Petra production facility is a mandatory to accomplish the demonstration project for the below reasons:

- a- most of the company current production is for ARTICLE 5 countries(84%) and most of these production percentage is for the gulf area which has high ambient conditions as the design ambient for this countries is about 46 c and a lot of designer consider it 50 c and if we compare the R410 a refrigerant pressure at this high ambient conditions and the r22 refrigerant we find that the pressure of r410a at this condition is about 550 psig while it is 320 psig using r22 refrigerant ( so the pressure difference is about 230 psig, R410 a is much higher pressure than r22 specially at this high ambient conditions )and the current tube thickness for the heat exchanger coils produced in petra company cannot be used in this case as it is unsafe to use it in this high pressure levels in this article 5 countries and specifically in these high ambient markets which is the major markets for Petra

So changing the tube thickness and outside diameter is not a luxury issue it is a mandatory issue to produce these kind of heat exchanger which is suitable for this R410 a refrigerant as it will be unsafe at all to use the current heat exchangers in this application and we see that this change is a vital change required to implement this demonstration project in the right and schedule time as agreed between all concerned parties

- b- The R410A is a high pressure refrigerant and it is also high GWP refrigerant And the requirement for supplying mechanically and electrically safe units is a major item which all the standards and specifications ask for, keeping the heat exchanger facility as is without changing the tooling will limit the use of this refrigerant (R410a) in the developing countries because simply it is not comply with pressure safety requirements and will lead the consultants and the certification bodies refuse to use this type of refrigerant in the A/C applications and they will prefer to use the r22 refrigerant is it is more safe if we keep the same tube thickness and OD in PETRA facility.

So Petra is recommending now to have a new machine (fin press + hairpin bender + tube benders +pipe cutting machines ) which can handle the production requirements for **1/4 inch OD** finned and tube heat exchanger as **Petra have only machines** which can produce **3/8 inch OD** WITH 0.3 MM TUBE THICKNESS).

- c- As per UL 1995 American standard which is used in many article 5 countries and as per attached table the minimum tube thickness is 0.6 mm for 3/8 copper pipes and the tubes in the heat exchanger coil must be capable to withstand a pressure of 5 times the normal

working pressure of the r410 a refrigerant at high ambient conditions in the most of article 5 countries while The heat exchanger tube thickness used in petra is about 0.3 mm thick with 3/8 inch OD this tube thickness is cannot withstand this test pressure as per UL1995 STD requirements so using higher thickness is a mandatory for refrigerants (R410A) Please see the attached documents from UL 1995 std which clarify the safety requirements for the minimum tube thickness and pressure tests requirements in this regards.

Recommendations:

We recommend to consider adopting new machines for heat exchanger production line and to consider this incremental operating cost for Petra project AND to adopt this decision in the 63 meeting

as this demonstration project cannot be completed 100% without adopting such changes in the heat exchanger production line of petra engineering ind.

**ATTACHMENT III**

**COMMENTS ON “A STUDY OF FIN AND TUBE COIL MANUFACTURING CHANGES FOR  
CONVERSION FROM R-22 TO R-410A” SUBMITTED BY THE WORLD BANK**

Comments on “A Study of Fin and Tube Coil Manufacturing Changes for  
Conversion from R-22 to R-410A”

**Standards and Tests**

Based on consultation with A/C manufacturers in Thailand, they considered two primary conditions that their products will encounter: one during shipment given Thailand’s hot climate condition, and another during normal operating condition.

During shipment, the temperature in a closed space of a delivery truck or container could rise significantly over the ambient temperature, as a result pressure inside A/C components could be much higher than those experienced during normal operating conditions and the refrigerant-containing parts could suddenly burst. To meet this stringent condition, the manufacturers carry out the pressure design of coils and other components in order to meet the burst test. The paper points out similar situation that could occur during shipment in a hot location like Arizona.

They have adopted UL 1995 Standards: Clause 61 (Strength Test) where all refrigerant-containing parts must be able to withstand 5 times the design pressure. While UL 1995 standards provide an alternative method: Clause 62 (Fatigue Test Analysis), to test samples under simulated normal operating conditions, the test sample is subjected to 3 times the design pressure which is lower than test pressure under the burst test. The manufacturers do not feel the fatigue test should be used as a substitute test on condition that could occur during shipment in hot climate condition.

**R-22 and R-410A coils**

Based on experience of major manufacturers, R-410A refrigeration system has a nearly identical overall system performance to R-22 due to its superior heat transfer performance despite lower COP. The paper claims that it is not necessary to made changes to the coil design when changing from R-22 to R-410A.

However, the pressure of R-410A system is nominally 60% to 70% higher than R-22, thus the R-410A coil must be redesigned in order to meet the burst test pressure. The paper claims that designing coils for R-410A to meet the burst test pressure would require the wall thickness to be 70% higher since the tube wall thickness increase proportionately with pressure. This would require additional copper and would be very costly.

The paper then suggests using a fatigue test as an alternative method to qualify R-410A coils without changing the tube diameter, thickness, nor the standard materials of construction.

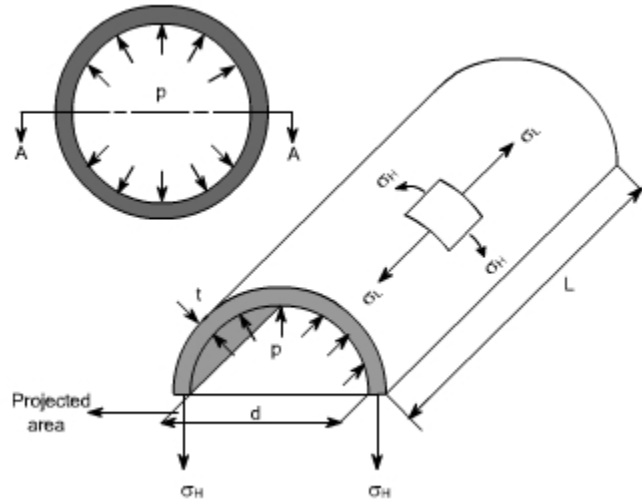
**WB comments:**

1. While we agree that fatigue test is a good method to test the system under simulated normal operating condition, we do not think it adequately addresses the above condition that could occur during shipment. If coils cannot withstand the typical burst test pressures needed to qualify R-410A coils, the recommendation should have been to find an engineering solution to solve the problem, instead of changing a testing method. Extending the result of the fatigue test analysis to

the case where the failure could arise from a different situation would not be technically appropriate.

2. On the tube wall thickness, we believe it is possible to design coil to meet the increase pressure without the abovementioned penalty and, in fact, that is how the industry is addressing the problem. We would like to use an example below to support our view:

Copper tube in condenser contains refrigerant at high pressure. The internal pressure causes three types of stresses on the tube: Circumferential or Hoop Stress ( $\sigma_H$ ), Longitudinal Stress ( $\sigma_L$ ), and Radial Stress ( $\sigma_R$ )



If the ratio of the thickness to the internal diameter is less than about 1/10 then it can be considered as thin-walled cylinder. It is assumed that the hoop and longitudinal stresses are constant across the thickness. It may also be assumed that the radial stress is small and can be neglected. Under thin-wall assumption, hoop stress can be calculated using the following equation:

$$\sigma_H = p \times \left( \frac{d_o}{2t} - 1 \right)$$

Longitudinal stress is represented as follows:

$$\sigma_L = \frac{1}{2} \left( p \times \left( \frac{d_o}{2t} - 1 \right) \right) = \frac{1}{2} \sigma_H$$

Where  $p$  = pressure,  $d_o$  = outside diameter, and  $t$  = thickness

A tube could fail when subjected to an excessively high internal pressure. Since hoop stress is twice as much as the longitudinal stress, it usually fails by bursting along a longitudinal direction.

R-22 baseline case:

|                         |        |
|-------------------------|--------|
| Outside diameter, $d_o$ | 7.0 Mm |
| Wall thickness, $t$     | 0.3 Mm |



|  |       |      |
|--|-------|------|
| Design pressure                              | 450   | Psig |
| Test pressure, $p$ (5 times design pressure) | 2,250 | psig |

Using the above equation, the hoop stress is calculated to be 24,000 psig at test pressure. This result hoop stress is about 80% of copper yield strength.

From the above equation, it is clear that the hoop stress can be reduced either by reducing the diameter, increasing the thickness, or both.

By setting 24,000 psig as the design limit, we can vary the diameter and wall thickness of the copper tube until we can find the combination that can be used to manufacture R-410A coil that meet the test pressure of 3,900 psig (5 times the design pressure at 780 psig).

Option 1: Fixed outside diameter at 7.0 mm, wall thickness varies from 0.3 to 0.5 mm

|                            |        |        |        |      |
|----------------------------|--------|--------|--------|------|
| Outside diameter ( $d_o$ ) | 7.0    | 7.0    | 7.0    | mm   |
| Wall thickness ( $t$ )     | 0.3    | 0.4    | 0.5    | mm   |
| Hoop stress                | 41,600 | 30,225 | 23,400 | psig |

The baseline copper tube (7.0 mm diameter and 0.3 mm wall thickness) would encounter hoop stress significantly beyond the design limit if used in a R-410A system. The wall thickness will need to be increased to 0.5 mm in order for the hoop stress to be within the design limit of 24,000 psig. This would result in additional copper requirement and not practical.

Option 2: Fixed outside diameter at 5.0 mm, wall thickness varies from 0.3 to 0.4 mm

|                            |        |        |      |
|----------------------------|--------|--------|------|
| Outside diameter ( $d_o$ ) | 5.0    | 5.0    | mm   |
| Wall thickness ( $t$ )     | 0.3    | 0.4    | mm   |
| Hoop stress                | 28,600 | 20,475 | psig |

In this case, the wall thickness will need to be increased to 0.4 mm in order for the hoop stress to be within the design limit.

While reducing the diameter of the tube and increasing wall thickness would result in decreasing flow area and increasing pressure drops. This could be offset by lower volumetric flow rate and better heat transfer and transport properties of R-410A.

## Conclusions

We would like to conclude that a simple engineering solution can be found to manufacture coil for R-410A system that meet the burst test requirement under UL1995. Reducing the diameter of the tube is required by the manufacturers in order to ensure adequate strength of the condenser without compromising its market competitiveness by avoiding excessive increase in the cost of coils.

Should you have any further comments or question, please do not hesitate to contact us.

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