



联 合 国  
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Distr.  
GENERAL  
UNEP/OzL.Pro/ExCom/65/51  
15 March 2012  
CHINESE  
ORIGINAL: ENGLISH

执行蒙特利尔议定书  
多边基金执行委员会  
第六十六次会议  
2012年4月16日至20日，蒙特利尔

与热交换机制造的工具更新改造相关的增量成本  
(依照第 61/45 号决定与第 62/61 号决定)

1. 秘书处在执行委员会第五十九次会议筹备阶段确定了与相对于增支经营费用的零部件制造转换费用有关的问题。执行委员会通过其第 59/14 号决定，要求将该问题的讨论延期至第六十次会议，后来又通过第 60/45 号决定，将其延期至第六十一次会议。第六十一次会议审议并通过了相关文件（UNEP/OzL.Pro/ExCom/61/51）中提出的问题，但本次会议没有讨论对热交换机制造技术转换项目供资的问题。

2. 执行委员会通过的第 61/45(c)号决定要求秘书处编制一份关于肋片管式热交换机制造技术与工具更新改造相关增支费用额度的问题，供执行委员会第六十二次会议审议。秘书处编制的文件（UNEP/OzL.Pro/ExCom/62/55）见本文件附件一。在执行委员会讨论期间，一位代表认为，这份文件未能充分考虑其中所涉及的高技术问题，因此，对文件的质量提出质疑。为此，执行委员会通过第 62/61 号决定，要求基金秘书处在广泛征求相关专家、执行机构和执行委员会委员意见的基础上，编制一份新的关于热交换机制造技术与工具更新改造相关增支费用的文件，以方便委员会审议。秘书处根据第 62/61 号决定，在充分顾及所提高度技术性问题的情况下，编制了 UNEP/OzL.Pro/ExCom/65/53 号文件。执行委员会开始对文件进行了审议，但经初步讨论后，同意将对与热交换机制造技术与工具更新改造相关的增量成本的讨论推迟到第六十六次会议进行。

3. 秘书处重刊了为第六十五次会议编制的文件，增加了世界银行就此问题提出的技术性评论。这些评论载于本文件的附件三。

## 方法

4. 秘书处对第六十二次会议的讨论进行了分析，并且详细记录了一位执行委员会委员质疑的、与技术问题相关的部分内容，从而为大量具体问题的调查提供更多细节。此外，秘书处还收到了空调设备制造技术向 HFC-410A 转换的示范项目受益机构（约旦佩特拉工程实业有限公司）研发部门的来信，其中关于资助热交换机技术转换项目的决定尚未获得通过。相关信息分别在载入本文件附件一和附件二。在印度尼西亚氟氯烃淘汰管理计划项下转换项目讨论期间，秘书处就同一问题与相关执行机构（尤其是开发计划署）进行了深入讨论。秘书处还分析了相关出资情况，并且编制了一份亟待审议的技术问题清单，最后得出结论认为，相关工作必须由一名外聘专家完成。秘书处确定了一名合适的专家并与其签约，聘请他开展相关研究工作。这名专家已于今年年初退休，是制冷技术方案委员会委员，曾是一家大型空调制造公司冷却装置开发工程经理、压缩机技术负责人。相关研究成果已载入本文件附件一中。

5. 本项技术研究工作重点关注从 HCFC-22 转换到 HFC-410A 期间，空调制造行业的作业规范。其中有些问题可能值得在本导言中提及：

- (a) 专家指出，系统压力与管壁厚度是否有必要增加的问题与产品在运输期间可能产生的高温和承受的相关压力有关，也就是说，与产品使用地点的气候无关，运输期间承受的压力水平高于运行时承受的压力水平；
- (b) 专家还指出，在任何情况下，加压阻力都可确定为上文（a）款所述温度定义的最大可能压力的倍数；
- (c) 他还指出，为了应对从 HCFC-22 到 HFC-410A 的转换所面临的挑战，全行业都应改进旨在确保加压阻力的方法，即从一种静态试验方法（试验压力预

计为最大压力的五倍）改为一种更加复杂的动态试验方法（更贴近实际），其中试验压力在最大可能压力的三倍以内，而系统则必须承受所施加的前 200,000 周最大可能的压力，然后再减压。专家解释说，尽管动态试验的最大压力低于静态试验，但是这种试验方法会给设备施加较大的额外应力，并且会呈现许多采用静态压力试验所无法看到的弱点；

- (d) 因此，空调系统（一般情况下）和热交换机（特殊情况下）的设计特征必须尽可能完美，必须将设计变更降至最低限度，但是不能出现任何需要做出重大调整的变更。

6. 为了确保产品质量，咨询专家还对与动态试验以及必要的补充工具相关的可能成本进行了量化分析。此外，相关研究工作还将详细解释一系列非常典型的设计变更，许多解释都是非常难得的实践经验，因此可作为执行机构和企业在进行技术转换时的指南，为再设计工作提供指导。秘书处认为，这项研究直接或间接涵盖了执行委员会内部讨论或与秘书处讨论时提出的全部技术问题。

7. 下文表 1 所示为热交换机技术转换的预估增量成本。

表 1：热交换机转换的成本

成本项目	中型企业的成本（美元）	小企业的成本（美元）
工具	15,000	15,000
试验设备	100,000	视具体情况而定
不可预见费用	11,500	视具体情况而定
各家企业的合计	126,500	≥15,000

8. 秘书处与咨询顾问讨论了与第 5 条国家生产商所需资助相关的研究可能得出的结论。它认为，一般而言，对于即将实施的工程，向 HFC-410A 的转换的要求更为迫切，究其原因，与其说是出于改变制造技术的需要，还不如说，考虑到一个循环周期内制冷剂不同的动态特征而采用较小的设计变更；不引入这些变更似乎会导致频繁的震动、造成接头断裂等问题。由于热交换机内部及周围的接头非常多，动态试验可保证热交换机的强度和全面生产质量。

9. 咨询顾问认为，中等规模的制造商（比如约旦佩特拉工程实业有限公司）本身可开展与动态检验相关的工作。这些中型企业的特点在于其 HCFC-22 消费量都在 100 公吨以上，而且都已在企业内部建立了相对独立的认证与质量控制团队。对于小型制造商，成立一个国家级检验试验中心、让这些小企业进入该中心获取相关专业技术、开展检验试验可能更具实质意义。咨询顾问也认为，这种集中化的方法更有价值，因为开展相关检验试验需要具备一定的能力。小型制造商无需开展大量的试验来证明相关专用设备适合这些用途，也不需要开展足够的试验来自主开发相关专门技术。因此，咨询顾问建议，对于自行生产热交换机的大中型制造商，可帮助它们购置试验设备，对其改进生产工具给予少量的资助。对于小企业比较多的国家，可资助其建立专门的检验中心，为小企业提供相关检验工具，而对于无法进入专门检验中心的小企业，可资助其购置小型试验设备或检验装置，让它们自身的能力得到发挥。

10. 咨询顾问指出，大容量、单制式热交换机的生产可能需要有不同的工具需求，具体取决于生产过程中使用的工具自身是否具备或缺乏精确适应性或灵活性。这一问题可作为本次研究的一部分加以解决，因为其在很大程度上取决于具体的安装。然而，从表面上看，大容量热交换机生产所用的生产设施并非归跨国企业所有的问题，只出现在中国的一些制造商中，在中国，2015年之前的供资水平已经商定。因此，这一特殊问题的解决并非如此迫切。

11. 执行委员会决定，在有多份报告提交的情况下，确定符合条件的热交换机技术转换的最大成本，但在向热交换机转换项目供资的共性问题最终得到解决之前必须保留供资额度。相关项目及其预留资金额度如下文表 2 所列。

表 2：氟氯烃淘汰管理计划项下尚未获得批准的热交换机转换供资项目

项目名称	执行机构	决定	经核准的费用 (美元)	申请用于热交换机转换 额外费用 (美元)
淘汰约旦佩特拉工程实业有限公司单制式空调设备生产中使用的 HCFC-22 和 HCFC-141b	工发组织	60/41*	2,167,033	737,000
印度尼西亚：氟氯烃淘汰管理计划（第一阶段）（空调行业计划）	环境规划署	64/42* *	1,995,519	52,800

\* 《决定》b 段：执行委员会决定审议向与肋片管式热交换机制造技术转换相关的费用项目供资的问题，审议 UNEP/OzL.Pro/ExCom/ 60/47 号文件中所包含的相关政策问题时，应考虑废除依照上文（a）款做出的供资决定。

\*\* 《决定》h 段：执行委员会决定要求开发计划署作为牵头执行机构，在执行委员会对热交换机制造技术转换项目供资事宜做出决定之前提交热交换机转换项目供资申请，作为对印度尼西亚氟氯烃淘汰管理计划第一次付款的组成部分之一，但条件是，必须依照该决定计算供资额度，而且计算得出的供资额度不得超过 52,800 美元的最大金额。

12. 与做出上述决定时的预期相反，本项研究得出的技术解决方案没有说明是否有必要重新装备相关工具。尽管事实上实现转换需要更多的工具，但咨询顾问只认为有必要建立相关检验设施。秘书处在本项研究基础上计算得出的、对两个相关项目的供资额度如下文表 3 所示。

表 3：热交换机转换项目建议的供资额度

项目名称	建议的供资额度 (美元)
淘汰约旦佩特拉工程实业有限公司单制式空调设备生产中使用的 HCFC-22 和 HCFC-141b	126,500
印度尼西亚：氟氯烃淘汰管理计划（第一阶段）（空调行业计划）	52,800

### 秘书处的建议

13. 基于上述讨论、本文件附件一所载 UNEP/OzL.Pro/ExCom/62/55 号文件的内容以及附录一所述研究得出的信息，谨建议执行委员会考虑：

- (a) 在空调产品制造商开始向本质上具有较大工作压力的替代技术转换以及该制造商自身拥有热交换机生产设施的情况下，与热交换机动态检验相关的支出将成为技术转换费用的一部分，以确保其具备足够耐压力；

- (b) 要求秘书处针对上述情况下，运用 UNEP/OzL.Pro/ExCom/66/51 号文件中提供的信息、相关技术研究成果以及任何其他可用的相关信息，确定这些费用符合条件的程度并且增加的额度；
- (c) 对于“淘汰约旦佩特拉工程实业有限公司单制式空调设备生产中使用的 HCFC-22 和 HCFC-141b”，依照第 60/41 号决定，核准额外供资 126,500 美元（外加支付给工发组织的 9,488 美元机构支助费用）申请，用于支付与热交换机生产和检验技术革新相关的费用；
- (d) 对于“印度尼西亚：氟氯烃淘汰管理计划（第一阶段）（空调行业计划）”，依照第 64/42 号决定，核准额外供资 52,800 美元（外加支付给开发计划署的 3,960 美元机构支助费用）申请，用于支付与热交换机生产和检验技术革新相关的费用；同时，要求秘书处在引入与印度尼西亚基准消费量的确定相关的任何变革项目时，更新与印度尼西亚达成的协定。

**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



联合国  
环境规划署



Distr.  
GENERAL  
UNEP/OzL.Pro/ExCom/62/55  
29 October 2010  
CHINESE  
ORIGINAL: ENGLISH

执行蒙特利尔议定书  
多边基金执行委员会  
第六十二次会议  
2010年11月29日至12月3日，蒙特利尔

与热交换机制造工具更新相关的增支费用  
(第 61/45 号决定)

1. 秘书处在执行委员会第五十九次会议筹备阶段确定了与相对于增支经营费用的零部件制造转换费用有关的问题。执行委员会通过其第 59/14 号决定，要求将该问题的讨论延期至第六十次会议，后来又通过第 60/45 号决定，将其延期至第六十一次会议。

2. 第六十一次会议审议并通过了相关文件（UNEP/OzL.Pro/ExCom/61/51）中提出的问题的，但本次会议没有讨论对热交换机制造技术转换项目供资的问题。因此，执行委员会通过第 61/45(c)号决定，要求秘书处根据 UNEP/OzL.Pro/ExCom/61/51 号文件的相关章节，编制一份关于肋片管式热交换机制造技术与工具更新改造相关增支费用额度的文件，供执行委员会第六十二次会议审议。

### 热交换机的增支费用

3. 作为项目提案审查工作的一部分，秘书处发现了一个问题：热交换机制造技术转换是否被视为增支费用。以下段落正是要解释这一问题的技术内容，以便执行委员会能够就这一事项做出知情决定。

4. 本文件所指的热交换机制造涉及制冷剂-空气换热机的制造。其构件通常是一些带有与之垂直的板状铝制肋片的铜管。

5. 肋片通过络合染料制成，利用多次击打、穿孔，形成肋片。铜管外径在最低限度上要小于肋片的孔径，这样肋片就能够比较容易地与一套铜管准确连接。通常情况下，铜管先拉直，然后弯曲成 U 形（“U 形管”），这样每个管子都能够两次穿过热交换机；一个热交换机可能有许多管子。这些管子在一个水平面上将大量肋片穿起来（多达数百个）。一旦所有的 U 形管都安装了肋片，将一根内径略大于铜管的带有精密走珠笔尖的杆子穿过每个管子，略微扩大管子内部及其外径，从而使得管子和肋片完全匹配。在高速生产过程中，所有管子都在同一时间膨胀。如果生产量较低，则有时会一次膨胀一根管子。这些热交换机被称为肋片管式热交换机。

6. 制冷剂-空气热交换机在制冷和空调系统中很常见，尤其是在这些系统大规模生产时很常见。大批量生产时，换热机的每个型号都进行优化并且向一个外部供应商购买，或者更常见的是优化制造商的型号范围，在当地生产。通常各种容量的机组使用相同的外管直径。这些换热机的设计或材料在采用 HCFC-22 技术和当前各种 HCFC-22 替代物之间（氨和二氧化碳除外）没有重大差别。

7. 根据秘书处所征求的技术专家的意见，由 HCFC-22 变更为 HFC-410A 或 HFC-32 时，从系统性能的角度看不必缩小铜管的直径；变更为 HFC-407C 和碳氢化合物 HC-290 和 HC-1270 时的情况也是如此。然而，必须微调管壁厚度，以便增强力量以应对更高的 HFC-410A 或 HFC-32 工作压力。这种办法需要的设备投资要小得多，因而很受初始转换制造商的偏爱。另一个替代办法是使用特定的更昂贵的铜材制成管子，从而使尺寸相同的铜管能够承受更大的压力，或者两者兼而有之。根据一些项目提案的要求，缩小管子的外径利用了 HFC-410A 的性能多于 HCFC-22 的特定好处，导致系统小型化及重量和成本减少。节约成本非常重要，有利于在生产费用大幅度降低的情况下生产更好的热交换机。秘书处收到的项目提案显示，HCFC-22 的增量节约率可能高于 2 美元/公斤。

8. 然而，如项目提案所示，缩小铜管外径的工具更新改造费用极高。属于二氧化碳系统的热交换机可能需要使用更小直径的管子，因为二氧化碳系统的工作压力很高，而且每单位体积的容量都不同。使用可燃性制冷剂（碳氢化合物以及在更小程度上为 HFC-152a



和 HFC-32) 的系统可利用直径更小的管子大幅度减少制冷剂充注量, 从而与现有氟氢烃系统相比, 能够在几乎没有附加安全要求的情况下, 在上述制冷设备中使用可燃性制冷剂。

9. 生产热交换机的机器在某种程度上是定制的, 尤其是外管尺寸。外管尺寸发生变化, 就必须更换设备, 尤其是用于制造肋片的染料、弯曲铜管的机器、自动焊接机和扩管所使用的机器。对于消费品(室内空调等)生产而言, 所有这些都通常是完全自动的专用精密机床, 其改造或更换成本相对较高。商业和工业产品的自动化制造程度往往更低, 便于客户定制。然而, 所需的机器也是高精度的机器。

10. 目前许多公司独立制造自己的热交换机, 使他们在根据客户的要求设计和制造大型空调系统时有更大的灵活性, 同时也可能在经营费用方面有所节约。在秘书处注意到的几乎所有情况下, 制冷和空调设备制造商都是独立生产自己的热交换机。

11. 第六十一次会议之后, 秘书处在向中国派遣代表团期间继续与各机构和产业代表进行商讨。秘书处文件的概念不断得到解释。尤其是有一个机构提及通过重新设计和优化热交换机来提高空调能效的可能性。产业代表谈到同行公司的一些实例; 在这些公司, 相关转换, 尤其是变更为 HFC-410A, 业已实现; 热交换机也进行了改造, 并且认为热交换机制造转换与生产线转换之间存在有机的联系。

12. 在上述讨论中, 秘书处特别谈到执行委员会的第 61/44 号决定, 该决定要求秘书处在评价制冷和空调部门氟氢烃转换项目的零部件更新情况时, 坚持既定做法, 从而在进行转换之后, 各零部件的界定特征在很大程度上保持不变, 或者在没有提供类似零部件时, 仅做实施转换所需的改良。秘书处没有质疑热交换机生产线转换是否与制冷剂转换有着有机的联系, 但是对缺乏技术需求和额外经营效益时是否需要多边基金为转换提供支持提出了疑问。虽然秘书处没有对一个执行机构关于能效的观点的技术内容提出异议, 但是以提高能效为唯一目的的活动不符合多边基金的要求。如果某个受益人希望转换制造技术以实现能效效益, 就必须提供配套资金或进行共同筹资。

13. 建议所有利益攸关方加入秘书处关于这方面的讨论, 针对变更管子外径提供书面的技术理由, 但不包括与提高能效或节约成本有关的理由。尽管明确提出了最后期限, 并且在一种情况下发出了催复通知, 但是秘书处尚未收到有关这方面的任何意见。

#### 秘书处的建议

14. 根据以上报告以及第 61/45 号决定, 秘书处向执行委员会提出了与提交第六十一次会议的 UNEP/OzL.Pro/ExCom/61/51 号文件所载相同的建议, 即执行委员会可能希望考虑, 在制冷或空调系统从氟氢烃向不易燃氢氟碳化物转换的情况下, 肋片管式热交换机内部变更管径与工具更新改造相关资本成本不视为增支费用, 因为这些被视为构成一项不可避免的技术更新。

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**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.

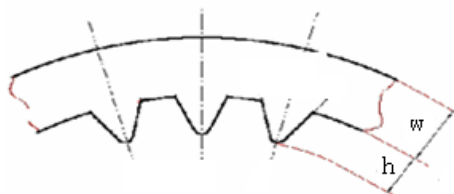
Since R-22 and R-410A have significantly different thermodynamic and physical properties, one might expect significant changes in coil design. It follows that manufacturing processes, capital equipment and tooling may need to be replaced or changed. This study will examine the two basic aspects of the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

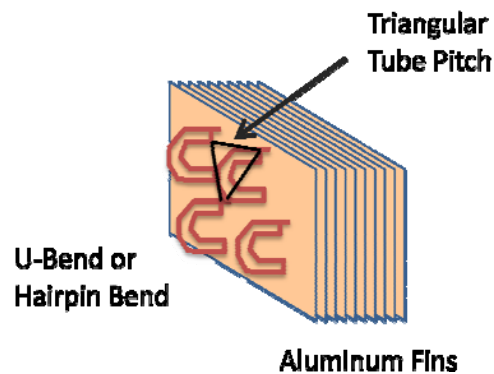
## 2. Baseline Coil Information

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><u>6.98 ++</u></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><u>650 ++</u></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>	<b>3300 ==</b>	<b>6.282 (-10%)==</b>	<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.

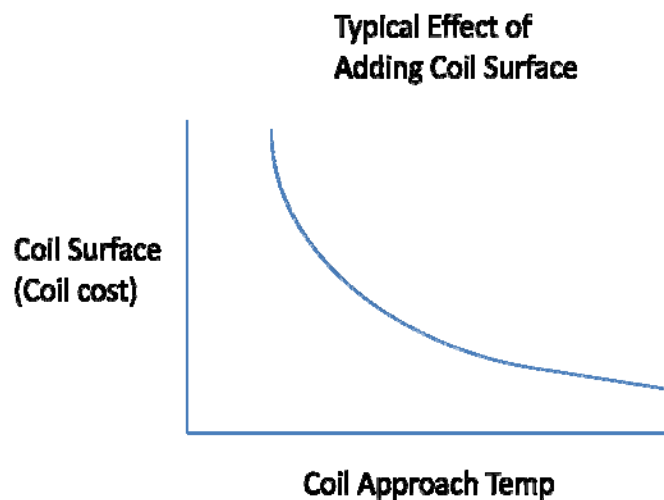
The difference between the saturated evaporating temperature and the saturated condensing temperature is the pressure difference or 'lift' that the compressor must work against. The smaller the difference is, the lower the power consumption.

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.”

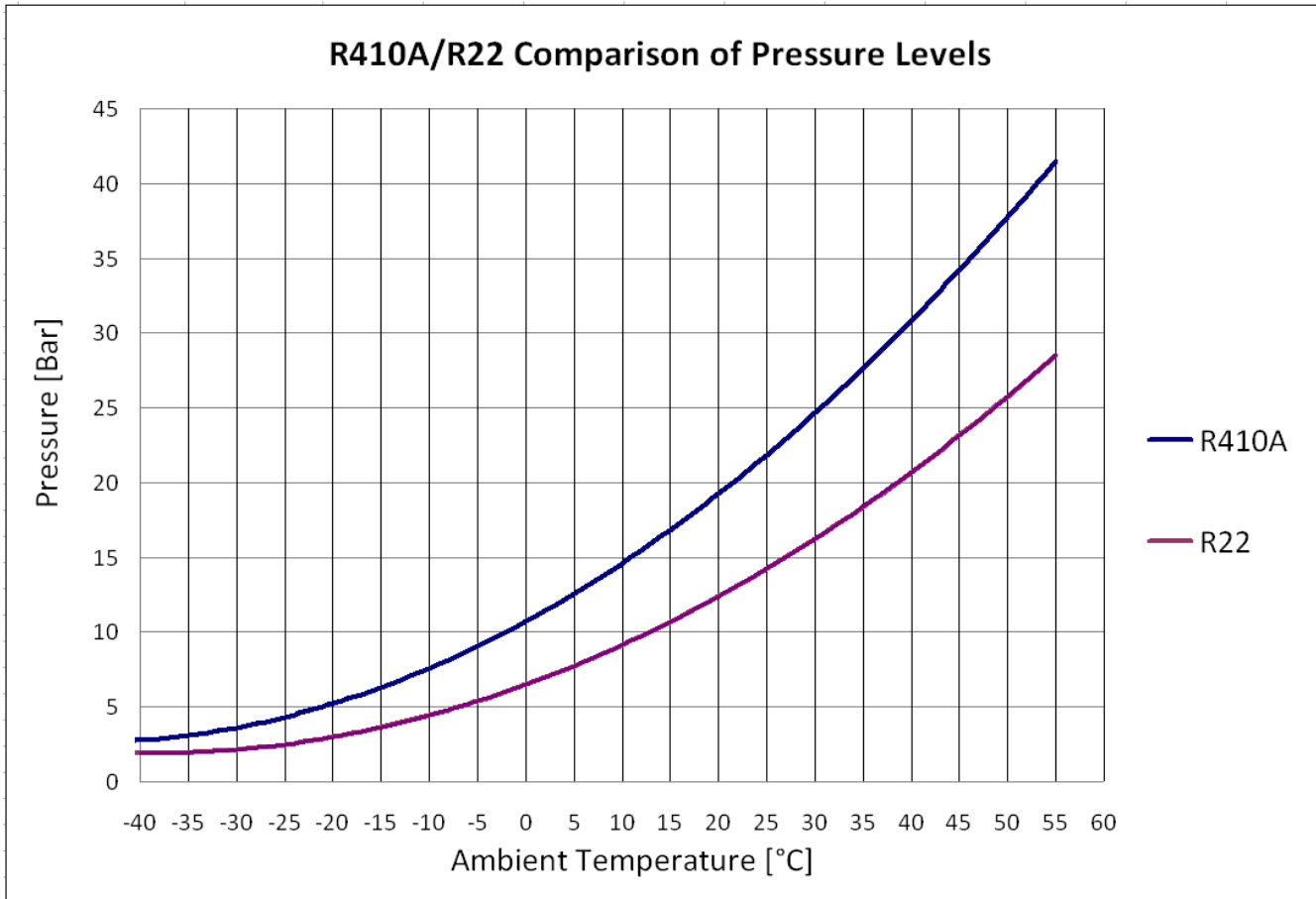
Annex II

**“ 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 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.

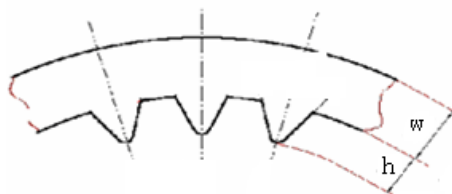
Since R-22 and R-410A have significantly different thermodynamic and physical properties, one might expect significant changes in coil design. It follows that manufacturing processes, capital equipment and tooling may need to be replaced or changed. This study will examine the two basic aspects of the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

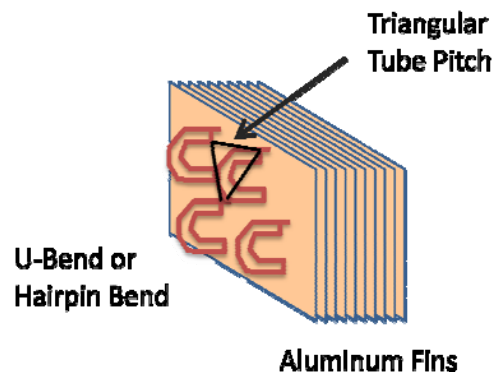
## 2. Baseline Coil Information

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><u>6.98 ++</u></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><u>650 ++</u></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>	<b>3300 ==</b>	<b>6.282 (-10%)==</b>	<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.

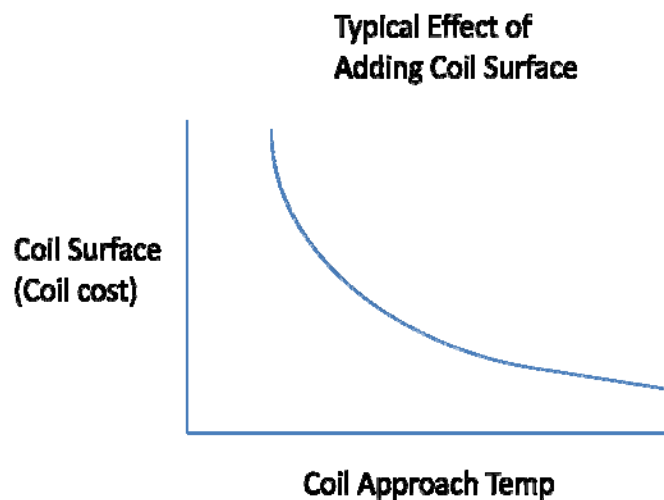
The difference between the saturated evaporating temperature and the saturated condensing temperature is the pressure difference or 'lift' that the compressor must work against. The smaller the difference is, the lower the power consumption.

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.”

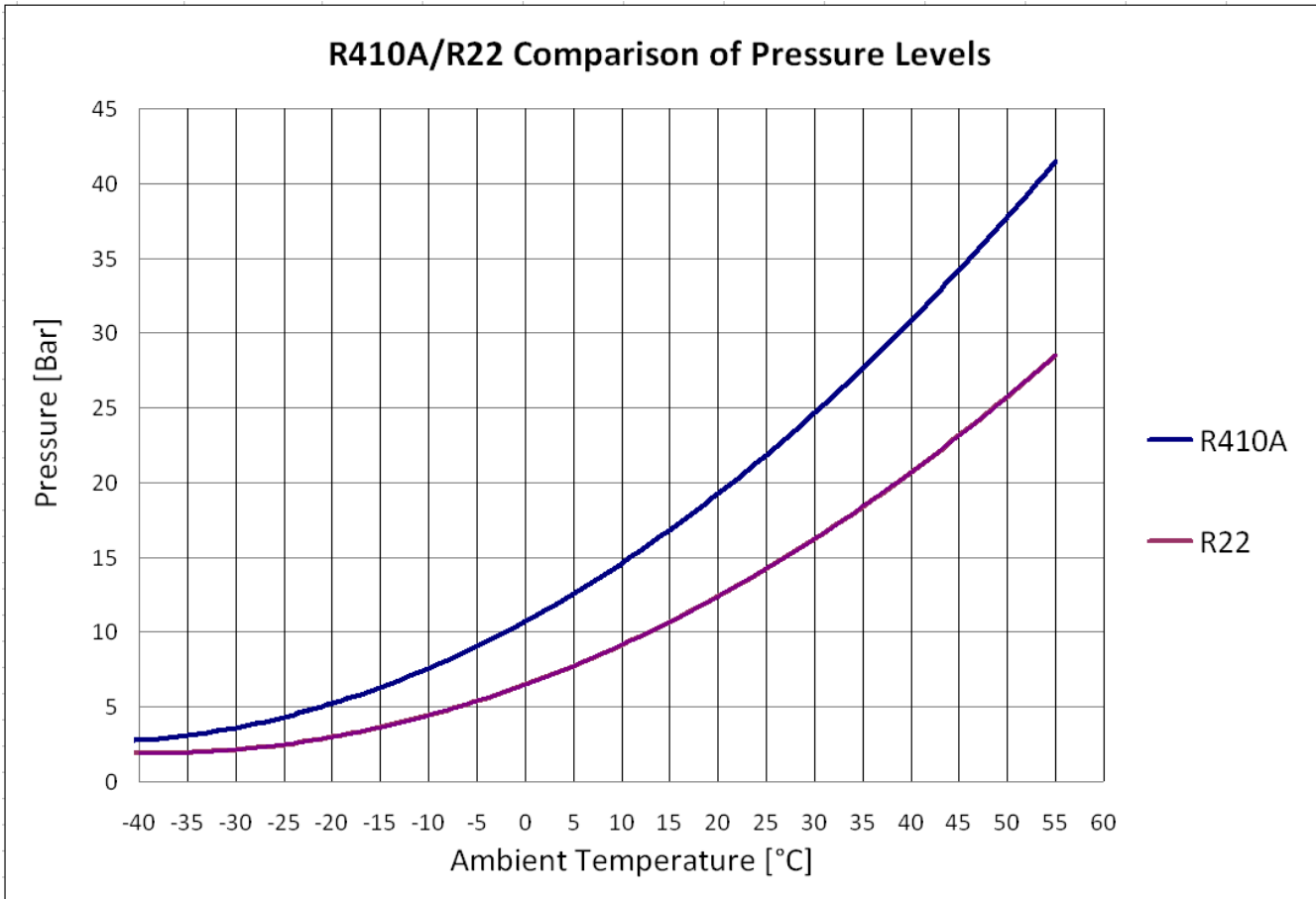


Annex II

**“ 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.

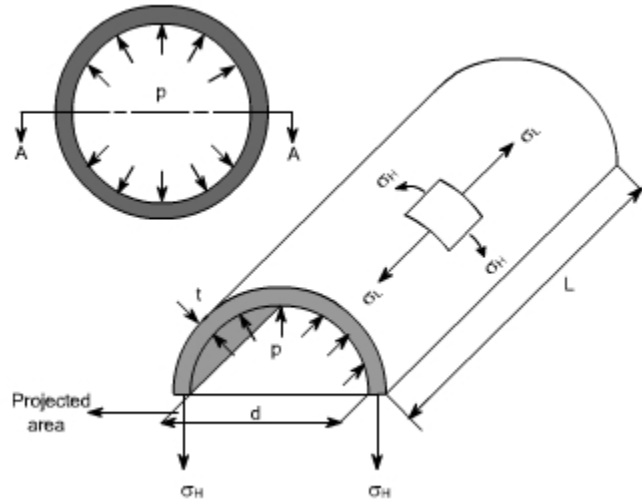
**WB comments:**

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.

MP Operations  
The World Bank  
6 March 2012